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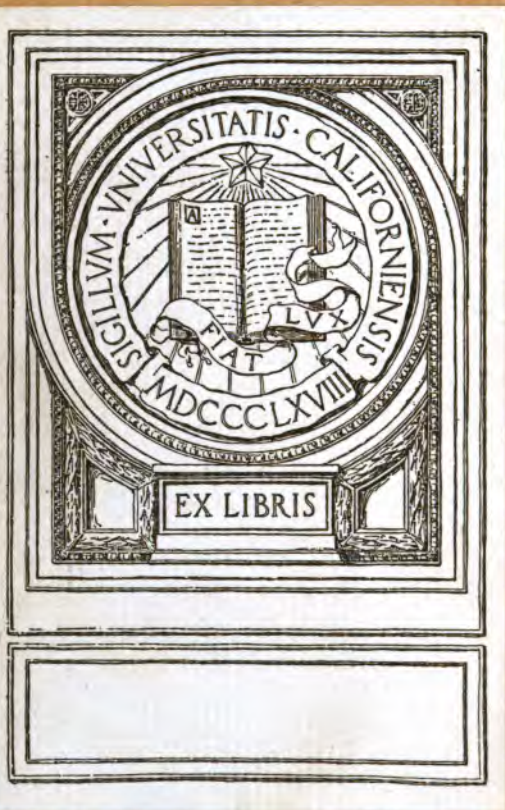
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# MECHANICAL LABORATORY METHODS

THE TESTING OF INSTRUMENTS AND MACHINES  
IN THE  
MECHANICAL ENGINEERING LABORATORY  
AND IN  
PRACTICE

  
BY

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**95 ILLUSTRATIONS**



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## PREFACE

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IN considering how best to describe what this book is intended to be, the author finds thoughts uppermost of what it is intended not to be. He has undertaken a subject already treated by a few admirable texts because there appeared in his own teaching the need of a *brief* work on the broad principles of experimental apparatus and methods. Therefore this book does not contain lengthy and detailed descriptions, nor does it attempt to cover all the instruments or methods of any class so that the student will understand them in all their details before commencing his work in the laboratory.

It is obvious that the average student can learn details of instruments and apparatus much more readily by actual examination of them than he can from printed and pictured descriptions. He should, of course, know the principles and purposes beforehand; then any deficiency of information may be made good by maker's pamphlets or by personal instruction in the laboratory.

In dealing with any class of instruments, therefore, the text of this book explains the principles of that class, or a subdivision of it, so that a brief explanation covers any specific instrument. Illustrations to help such explanations are diagrammatic, and not detailed drawings, the purpose being to make a simple presentation.

Instructions on how to operate engines, boilers, etc., have been omitted. The practising engineer, presumably, does not need them. The student rarely memorizes them from a book, but he does grasp and remember them if imparted at the time

when they may be associated with the practice. It is believed that it is essential in any case to give such instructions personally in the laboratory.

The principles of all except a few types of engines, descriptions of engines of differing types, and topics such as thermodynamics, etc., have not been included. It is assumed that the reader knows how the usual engines work before he is required to test them; at all events, if he does not, there are many texts open to him on these subjects. Ordinarily the college student of mechanical engineering has received a working knowledge of them from his other courses. This book is therefore on the subject of testing exclusively.

The standards of the codes of the A.S.M.E. have been adopted in almost all instances, even where the author preferred other standards, for the sake of uniformity. The codes have not been quoted at length, however, because, in the first place, they are accessible to anyone, and secondly, their numerous items and many different expressions for practically the same results confuse, rather than help, the student.

The value of such a book as this is intended to be depends largely upon the clarity of its presentation. The author has had exceptional opportunities in this particular, through numerous rearrangements of the subjects dealt with; the object being always to make them most easily understood by the student.

The two main divisions deal with the testing of instruments and of machines, respectively. The first of these is subdivided so as to deal separately with a group of instruments of a single type for a given purpose. Under each sub-division the principles of the instruments considered are first related, and then are given the methods of testing them. A number of problems are included, by the working of which the student may familiarize himself with the principles of the instrument before experimenting with it.

The main division dealing with the testing of machines is similar in treatment; the principles introducing each subdivision in this case applying to the quantities usually sought in a test of the machine considered.

The table of contents may be used as a synopsis of experiments which may be referred to for the quantities to be measured in the test of a given instrument or machine.

A number of the testing methods described have not appeared elsewhere in book form. For some, the author is indebted to his former association with the University of Pennsylvania; for example, the method of setting the slide valve by the Bilgram diagram, the application of the method of least squares to indicator spring calibration, etc. Other of the methods, the author has developed at Syracuse University; notably those of testing indicator drum motion and reducing wheels. A number of original formulas are presented for the simplification of calculations from test data. Upon such depend the methods given for testing gas engines and producers. Among others may be mentioned the formulas for the products of combustion of fuels, those for simplifying indicated horse-power determinations, etc. The sections dealing with these subjects, in somewhat different form, have been previously published in *Power*. Thanks are due to its editor, Mr. Fred. R. Low, for his courtesy in allowing their use here.

To proportion such a work as this aims to be is difficult. One cannot help but be influenced by his personal inclinations toward one branch or another. Further, the field is too vast for one man to have covered thoroughly in all its branches through personal experience. In view of these facts, the author states, in conclusion, that he would appreciate information concerning errors or defects from any who may use the book.

JULIAN C. SMALLWOOD.

SYRACUSE UNIVERSITY, January, 1914.



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# MECHANICAL LABORATORY METHODS

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## INTRODUCTION

### THE PRINCIPLES OF EXPERIMENTAL MEASUREMENTS

THE science of experimental engineering rests primarily upon the art of measurement. Conclusions relating to general laws or specific operating conditions, formed from test results, stand or fall according to the accuracy of measurement.

**Absolute Accuracy.** There is no such thing as absolute accuracy of measurement except by accident. For instance, the length of a bar, measured with a scale graduated to one-hundredths of an inch, may appear to be 3 ins. By chance, it may be that the bar is 3 ins. long exactly, neither more nor less by an infinitesimal fraction. This, however, is very improbable. A micrometer may show it to be 3.005 ins. long. An instrument of still nicer capability might yield another decimal place, and so on. Thus, absolute accuracy presupposes an instrument with graduations corresponding to infinitely small fractions of a unit.

Assuming that this condition could be sensibly complied with, it would still be necessary to prove the correctness of the instrument within its own graduations, and this could only be done by comparison with absolute standard lengths, that is, material objects known to have a definite linear dimension. But this implies measurement by an absolutely accurate instru-

ment. Thus, to create such an instrument, it is necessary to have the very article desired.

**Accidental Errors.** Aside from instrumental limitations, accidental causes of error are a sufficient bar to absolute accuracy. Accidental errors are all those that cannot be eliminated by instrumental or other corrections. They are due entirely to chance and are not systematic except according to the law of probability and chance. Such errors are incurred in the calipering of a bar, for example, when the calipers are held on a slant instead of on a true diameter, when the calipers are held with varying degrees of pressure, when temperature changes momentarily affect the instrument, and so forth.

Accidental errors are equally likely to be positive or negative from which it follows that in a series of observations upon the same quantity the probabilities are that the errors of excess will equal those of deficiency.

**Personal Errors.** There is another class of errors known as "personal errors" whereby a particular observer, by reason of his character, habitually reads too high or too low. This error may be estimated scientifically, the result being what is known as the "personal equation." Generally, in mechanical experimentation, this quantity is not an important one, being small in comparison with the usually unavoidable errors.

We have, then, to consider two classes of errors which are bars to absolute accuracy in any measurement, namely, instrumental and accidental. From an engineering standpoint, however, absolute accuracy is neither necessary nor desirable, for it would be too laborious and cumbersome. It is not worth while to produce a result with ten or twenty significant figures when three or four are enough for practical requirements. Only that degree of accuracy is sought which accords with the dictates of common sense.

**Per Cent of Error.** When a result is represented by three significant figures the error occurring through the omission of

the fourth is less than one half of one per cent. Thus, the quantities 1010, 101.0, etc., may be correctly expressed 1015, 101.5, etc., but in each case the error is less than 5 parts in 1000 or one half of one per cent. With the higher digits the per cent error is correspondingly less, as when 999.0 is used for 999.5, the error then being about 0.05 per cent. In most cases we cannot secure results with less than 1 per cent of error, so that three significant figures are ample for their presentation. In many cases even less accuracy than this is consistent with the object or the conditions of the test.

**The Precision and Accuracy of Instruments** should be examined before using them for experimental measurements. Precision has reference to the fineness of graduations, accuracy to their correctness. The value of the smallest graduation of an instrument is called its least count. Suppose, as an illustration, we wished to weigh separately a number of objects of approximately 100 lbs. so as to secure less than 1 per cent of error, that is, an error of less than 1 lb. A scales with a least count of 2 lbs., would do since a half a graduation could be estimated by eye with less probable error than 1 lb. A scales of any less precision however, would not be adequate.

**Standards.** The accuracy of an instrument is established only by comparison with some "standard unit," itself a copy, or a copy of a copy, of the "primary standard" kept at Washington. The details of this sort of testing are given in Part I, but it is well to emphasize here that the necessary closeness of the approximation of any standard used to the primary standard depends entirely upon the least count of the instrument to be tested. Generally the standard is sufficiently accurate if it is correct within one quarter of this least count, and, if the standard is an instrument, its least count need not be less than one-half that of the instrument tested. It is thus seen that the word standard must be accepted with a relative sense, and that for engineering purposes secondary standards may be improvised

which are just as useful as the most accurate standard obtainable.

Assuming instrumental correctness as great as consistent with the least count, there are two ways of securing greater accuracy of measured results: first, by using an instrument of finer graduations, and second, by making numerous repetitions of the measurement. When the latter is done a result with less probable error is obtained from the average of the observations than from any one of them. This is because by so doing the accidental errors previously mentioned are to some extent eliminated.

**Intrinsic Evidence of Accuracy.** In experimental engineering one of the greatest aids to accuracy is repetition. It is only by repeated trials that accurate conclusions can be framed because, not only are there variations due to erroneous measurement, but the quantities measured are generally variable ones, themselves subject to chance. Under such circumstances, a single determination proves nothing and indicates little. On the other hand a series of determinations not only reduces the probable error of the result, but furnishes intrinsic evidence of its value. Consider, for example, a series of four measurements made upon a single constant quantity by an observer, A. These are 103, 98, 101, 98, the average of which is 100. Assuming 100 to be the correct result (it is the best obtainable consistent with the observations, although, of course, not correct) then the error of each observation is the difference between it and the mean, or +3, -2, +1, -2, the plus and minus signs indicating whether the errors make the individual determinations greater or less than the mean. Now, if another observer measures the same quantity with the results 101, 100, 102, 101, averaging 101, the errors of his determinations will be 0, -1, +1, 0. It would be concluded at once that B's measurements and result of 101 were more accurate than A's because his errors, figured from the mean, are less than A's.



**Compound Quantities.** Most engineering measurements are upon compound quantities, the components of which are variable. The following parallel illustrates this case and also the graphic method later to be described. Fig. 1 represents the floor of a room; and the solid diagonal line, the path of a ball that rolls across it. It is desired to locate as exactly as may be the path as it is traversed by the ball. The only exact data we have are that it is a straight line and starts at 0. At a given instant, one observer measures the distance of the ball from the wall  $OX$  and another from  $OY$ . These distances,  $x$  and  $y$ , are sufficient to locate one of the positions of the ball and therefore

its path, provided they are accurately determined. Accidental errors prevent this, however, so that the path is falsely located on the line  $Oa$ . Obviously, if a number of points, instead of only one, could be located, a better result would ensue.  $b$ ,  $c$ , and  $d$  are such points. It is then found that these points do not lie on the

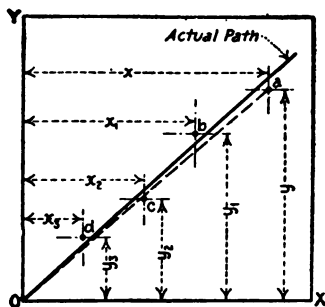


FIG. 1.

same straight line, the distances from the true path being the errors of their determinations. The question now arises how should these discordant observations be made to agree upon one result, which, though perhaps not the true one, is in best accord with the data. It is possible to locate a different line with each pair of observations,  $x \div y$  locating one,  $x_1 \div y_1$ , another, and so on. One might conclude that the average value of  $x \div y$  would be best, but it can be shown by the theory of errors that the arithmetical mean does not yield the best result upon indirectly determined quantities. In this case, the result is obtained by measuring two other quantities; the determination

is therefore indirect. The best result (referred to generally as "most probable") is a line so located that the sum of the squares of the distances from it of the points  $a, b, c$ , and  $d$  shall be a minimum. Such a line can be determined mathematically, but it is generally drawn by eye judgment.

**Variables, Independent and Dependent.** A large part of mechanical experimentation is parallel to this simple illustration. Two variables, bound together by a more or less rigid law, are measured, and from the result the law is deduced. The law is not always represented by a straight line, but often by a curve. Whichever it is, it is the business of experimentation to find.

One of the variables can always be controlled; the other then follows it according to the law binding them. For example, when a spring is extended by a force, there are two quantities, namely, the force and the extension, which combined give the law of the spring. In their measurement we may add predetermined increments of weight and let the extension vary as it may, or we may add weight enough to cause predetermined increments

of extension and let the weight increase as it may. The predetermined quantity is called the "independent variable"; the other, "dependent."

It is almost always desirable to present such measurements as points and to locate by them a smooth curve. This is done by adopting a pair of rectangular axes,  $OX$  and  $OY$  and scaling them according to the units measured. In Fig. 2, for instance, 1 in. along  $OX$  represents 10 lbs. applied to a spring and 1 in. along  $OY$  represents  $\frac{1}{4}$  in. of

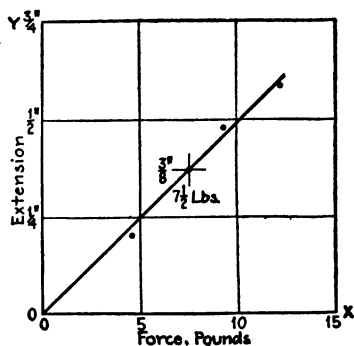


FIG. 2.

lbs. applied to a spring and 1 in. along  $OY$  represents  $\frac{1}{4}$  in. of

spring extension. The observations 7.5 lbs. and 0.375 in. are thus shown by the point indicated, and so on.

In many cases there are three variables, as the pressure, temperature and volume of a gas. It is then customary to keep one of them constant throughout a test, arbitrarily control another, constituting the independent variable, and let the third vary as it may.

If the relation between the variables appears to be satisfied by some other curve than a straight line, such a one is drawn.

**Intrinsic Evidence of Accuracy.** If any point of a plotted series lies markedly off the line, it may be concluded logically that its determination contained some large error, or mistake, and it may be thrown out of consideration entirely. Similarly, the value of a set of measurements may be judged by the closeness of the plotted results to a smooth curve.

It will be readily seen that the value of graphic presentation of results lies in the fact that their relative variation is visualized and can therefore be grasped by the mind much more easily than could mere columns of figures.

**Conventions for Plotting.** The independent variable, as a matter of convention, is plotted horizontally, the other vertically. Exceptions to this rule are calibration curves and stress-strain diagrams of materials; the one having instrument readings, the other, deformations, plotted as abscissas.

**Compound Variable Quantities.** In the case of the spring the two variables, when combined, give a constant quantity, namely, the number of pounds per inch of extension. It is not often that the required compound quantity is a constant one, although it is sought to keep it so throughout one set of measurements. As an example, the mechanical efficiency of a steam engine depends upon the amount of work it is doing; the greater its output, within its capacity, the greater its efficiency. Suppose it is desired to measure the efficiency at a given output. The engine is operated to deliver this horse-power, no more and

no less. But, owing to variations in steam pressure, governor action, and the like, it is possible to keep the horse-power output only approximately constant; it fluctuates somewhat through accidental causes more or less outside our control. Further, even if it were possible to keep it constant, the internal friction of the engine varies and this causes the work done in the cylinder to vary. Consequently, the efficiency, which is the delivered horse-power divided by the power developed in the cylinder, will vary. So aside from variations in the results produced by unavoidable errors of measurement, there are accidental variations in the true value of the result, even when all the controllable conditions are kept as constant as possible. It is like measuring the height of a small spot of light which persists in fluctuating up and down through a limited space. Clearly, a single measurement would be altogether insufficient, for it might represent a high or a low value, even if accurately made. Obviously, the best that can be done is to keep controllable conditions as constant as possible while taking not one but a number of measurements. It may now be understood how such conditions will determine the precision necessary, because of the uncertainty of the true value of the result.

**Rules for Testing.** The following rules should be observed when measuring variable compound quantities.

*First.* For one value of the independent variable, the series of measurements made of each separate quantity may be averaged. All such averages may be used in a single calculation of the desired result.

*Second.* Observations of fluctuating quantities should be made at equal time intervals to secure a true average.

*Third.* The number of observations necessary of any quantity depends upon the constancy of the quantity.

*Fourth.* Tests of time rates should cover a sufficient duration of time to reduce the error of initial and final measurements to 1 per cent of the total quantity.

*Fifth.* In time rate tests, intermediate time readings should be taken to check rate constancy.

**Constancy of Conditions.** Referring to the third rule, if all the conditions of a test could be kept absolutely constant, only two sets of readings would be necessary. The two would give identical results; the second would be made only to check the accuracy of measurement of the first. In some cases this condition is approximated, as, for instance, electric motor driven blowers. In other cases some of the quantities fluctuate more than the rest; under these circumstances the time interval between measurements of these quantities should be smaller than for the others, so that there may be more values obtained for them.

**Duration of Time Rate Tests.** The fourth rule may be illustrated as follows. In measuring the pounds of water per minute flowing into a tank, the cross-section of the tank is measured, and the difference of water in it before and after an observed time interval is figured by noting the corresponding difference of the water levels. The water level is measured with a scale graduated to tenths of an inch. With such a scale, it is possible to make an error of  $\frac{1}{10}$  in. (half a division) at each measurement, totaling  $\frac{1}{10}$  in. It would be necessary, then, to make the duration of test long enough that  $\frac{1}{10}$  in. be 1 per cent of the total rise in water level, that is, long enough for the water to rise  $0.1 \text{ in.} \div 0.01 \text{ per cent} = 10 \text{ ins.}$  This calculation allows for the maximum error probable, but it is on the safe side.

**Intermediate Readings.** The table of observations on p. 10 illustrates the fifth rule. If the column of differences shows approximately equal values corresponding to equal time intervals, the result is valid. Any marked departure from constancy should throw suspicion upon the result. If the readings at 3:00 and 3:10 only had been taken, we should be ignorant as to rate constancy. As has been pointed out, the reliability of the result depends upon constancy of conditions.

Time	Height	Differences
3:00	5.0 in .....	
		0.95 in.
3:02	5.95.....	
		1.05
3:04	7.00.....	
		1.05
3:06	8.05.....	
		1.00
3:08	9.05.....	
		1.05
3:10	10.10.....	

A further advantage from intermediate readings is that they strengthen the determination against mistakes. This was shown in the illustration of the path of the ball. From the observations just cited, the result ordinarily would be figured thus

$$\frac{(10.1 - 5.0) \times \text{area}}{10 \text{ min.}} = \text{cubic inches per min.}$$

which is the same as though no intermediate readings had been taken. If, however, a mistake in measurement were incurred in the first or the last reading, as 6.0 instead of 5.0 ins., or 20.1 instead of 10.1 ins., this mistake would not be apparent, and the result misleading or worthless. Intermediate readings would disclose such a mistake, and the set of observations involved could be discarded without sacrificing the others.

In time quantity measurements it is always best to record the time of day instead of time intervals merely. This insures a correct record of the time. Otherwise it is easy to note an interval such as three minutes when two or four actually have elapsed. A further advantage is that possible irregularities in results may then be accounted for by related happenings that might be associated only by a knowledge of the time of occurrence.

Before closing the general subject, it is well to emphasize the importance of figuring in the laboratory rough results from the observations as soon as obtained. This applies to commercial as well as student work. By so doing, many faults in operation and in the application of instruments may be detected and remedied in time. It is especially valuable to plot a curve of the variables as the test progresses, for it will reveal the accuracy or error of the determinations at the time when such knowledge is most valuable.

# PART ONE

## THE TESTING OF INSTRUMENTS

### WEIGHTS AND FORCES

THESE are measured by comparison with known weights with the aid of a system of levers such as in a beam balance, or by reference to the amount they deform some elastic object, as a spring, which has been previously calibrated against standard weights. When a leverage system is used so that a large force may be measured by comparison with a small known weight,

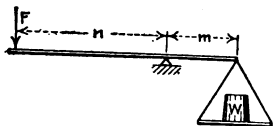


FIG. 3.

it is necessary to know the ratio of the lever arms. In Fig. 3, for instance, the force  $F$  is measured by the relation

$$W = \frac{n}{m} F$$

in which  $\frac{n}{m}$  may be referred to as the "leverage ratio."

The calibration of any force measuring apparatus rests primarily upon the force of gravity as a standard. "Standard weight" is a misnomer in that the *weight* of the body so called, being the



force of gravity between it and the earth, actually varies with the locality of the body. Standard mass is a better term. A standard mass establishes a standard force when the acceleration of gravity at the given locality is known.

### 1. CALIBRATION OF PLATFORM SCALES

**Principles.** The platform scales Fig. 4, is arranged so that a large weight,  $W$  to be measured, may be balanced by a small

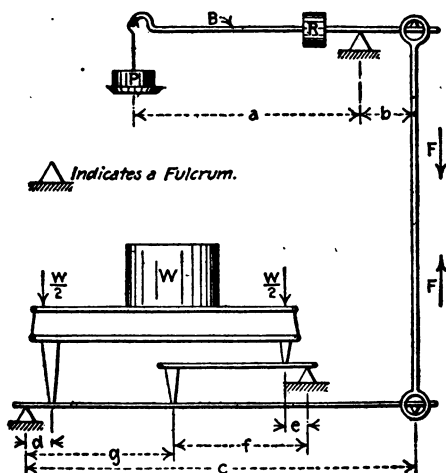


FIG. 4.—Platform Scales.

weight on the beam,  $B$ . The small weight is either or both  $P$  on the poise or  $R$ , the rider, acting at a variable distance from the beam fulcrum. If  $P$  balances  $W$ , then by the principle of moments,

$$P \times \frac{a}{b} = F = \frac{W}{2} \times \frac{d}{c} + \frac{W}{2} \times \frac{e}{f} \times \frac{g}{c}$$

It is arranged that  $\frac{d}{c} = \frac{e}{f} \times \frac{g}{c}$  so that

$$P \times \frac{a}{b} = W \times \frac{d}{c}$$

and

$$\frac{W}{P} = \frac{ac}{bd} = \text{the leverage ratio.}$$

**(a) The Sensitiveness of the Scales.** This is determined by placing a large weight on the platform, weighing it, and then finding the smallest additional weight that will cause a deflection of the beam which can be nicely balanced by the rider. This additional weight is also a measure of the precision of the scales, provided the beam graduations are small enough to take cognizance of it.

**(b) Beam Calibration.** Readings of the beam are taken corresponding to a number of standard weights. These should be sufficient to cover fairly the range of the beam. If standard weights of convenient size are not available, a number of small weights may be standardized for the purpose by using a calibrated scales with slightly greater precision than that of the scales to be tested. (See p. 3.)

If the instrument readings do not agree with the true weights a calibration curve should be plotted, having the instrument readings as abscissas and the true weights as ordinates. This curve may be used to get corrected values at any part of the beam.

**(c) The Leverage Ratio** may be found by measurement of the lever arms, but this method is difficult and subject to error. A better one consists of balancing a standard weight on the poise with a standard weight on the platform, then calculating the ratio of these weights. The nominal leverage ratio may be learned by examination of the poise weights. They are marked with their actual weights and the weights they are intended to balance, as, for instance, 1 lb.-100 lbs. This gives a nominal leverage

ratio of 100. To test the accuracy of this, place, for example, a standard  $\frac{1}{2}$ -lb. weight on the poise and a standard 50-lb. weight on the platform. If they do not balance, add enough weight (shot is convenient) to either one or the other until a balance is secured. This additional weight may then be accurately measured and the true leverage ratio found.

**(d) Poise Weight Calibration.** The indications of the poise weights are accurate if the leverage ratio is true and if the actual weights of the poise weights are as marked. They should therefore be weighed by a calibrated scales of sufficient precision. It should be noted that any error in the poise weight is multiplied in the instrument reading by the leverage ratio. The standard scales should therefore weigh these weights to within an amount equal to the precision of the scales to be tested divided by its leverage ratio.

If the actual weight of a poise weight is not as marked, or if the leverage ratio is not true, then the weight balanced on the platform is

Actual weight of poise weight  $\times$  actual leverage ratio

instead of the amount indicated by the marking. For instance, if the poise weight intended to measure 100 lbs. actually weighs 0.99 lb. instead of 1 lb., and if the leverage ratio is 99.5 instead of 100, then the weight on the platform balanced by it is  $0.99 \times 99.5 = 98.5$  instead of 100 lbs. In this way the true weight balanced by each poise weight may be found.

**Problem 1<sub>1</sub>.** Prove that it makes no difference in the instrument indications where  $W$  is placed on the platform.

**Problem 1<sub>2</sub>.** If the rider,  $R$ , is too light, will the resulting error be constant at all indications of the beam or will it vary and why? Will the error be plus or minus?

**Problem 1<sub>3</sub>.** If the poise is too light to bring the beam down with nothing on the platform, can a scales, otherwise accurate, be used for correct results without previous calibration, and how? If too heavy?

**Problem 1<sub>4</sub>.** Calculate the proper weight of the rider and check by weighing it.

**Problem 1.** If the poise weight marked 1 lb.-100 lbs. is verified to within  $\frac{1}{8}$  oz. of probable error, what would be the probable error of the instrument reading and the per cent error? *Ans.*,  $\pm 0.39$  lbs.;  $\pm 0.39\%$ .

**Problem 1.** Examine a scales to find if the leverage ratio may be adjusted.

## 2. EVALUATION OF A SPRING

**Principles.** In a large class of instruments, it is necessary to know the amount of force required to extend, compress, or twist a spring per unit of deformation. This quantity is called the "spring scale." According to Hooke's law, it is a constant within the elastic limit of the material. In many cases the movable end of the spring is attached to some combination of links and gears designed to indicate, magnify, or translate the motion. When the applied force is increasing, the friction of such links or gears makes the instrument read low, since the indicator is moving up the scale and friction tends to hold it back. With a decreasing force, the instrument reads high for a similar reason. Suppose an external force  $F$  is applied to a spring instrument, which is correct except for the effect of friction, first by increasing the external force to the value  $F$ , and then by decreasing it to this value. Then, if  $S$  is the spring scale,  $E_1$  and  $E_2$ , the extensions in the two cases, and  $X$  the friction,

$$\begin{aligned} F &= S \times E_1 + X \\ F &= S \times E_2 - X \\ 2F &= S(E_1 + E_2) \\ F &= \frac{S(E_1 + E_2)}{2} \end{aligned}$$

That is, the effect of friction may be eliminated by taking increasing and decreasing readings at each load applied, dividing the sum of the extensions thus found by two, and using the result as the extension caused by the external force.

(a) **Spring Scale by Graphic Method.** Take as an example the following measurements of force and extension.

Force lbs.	Extensions		
	Increasing In.	Decreasing In.	Combined In.
10	0.32	0.34	0.33
20	0.64	0.70	0.67
30	1.02	1.08	1.05
40	1.32	1.36	1.34
50	1.68	1.70	1.69

These are plotted as points on coordinating paper, and then is drawn what appears to the eye as the best straight line to satisfy each of the three sets of points.\* Fig. 5 shows the line for the increasing readings; the other two curves are similar. When it is desired to apply a calibration only to increasing readings, the ascending line is used, and similarly with the descending. The combined line gives the best results when the instrument is used for measurement of a fluctuating quantity.

To figure one of the spring scales, as for instance the ascending, the line for which is shown by Fig. 5, a point is selected near its extremity, as  $F_1 - E_1$ , this point not necessarily representing a pair of observations. It should, however lie on the line. Then the spring scale equals

$$S_{\text{(ascending)}} = \frac{F_1 - k}{E_1}$$

or, using numerical values,

$$= \frac{47.4 - 1.0}{1.60} = 29.0 \text{ lbs. per inch}$$

\* Certain researches of the author indicate that the decreasing values of force and extension do not follow Hooke's law within appreciable amounts and that, theoretically, they must follow a curved line. (Physical Review, October, 1911.) To represent them by a straight line is therefore merely a convenient approximation.

to be used when the readings of the instrument increase; similarly with the other two spring scales.

Note that the intercept,  $k$ , is taken into account. If the curve does not pass through the origin it is because of friction or backlash of the indicating mechanism or to a false zero reading of the extension.

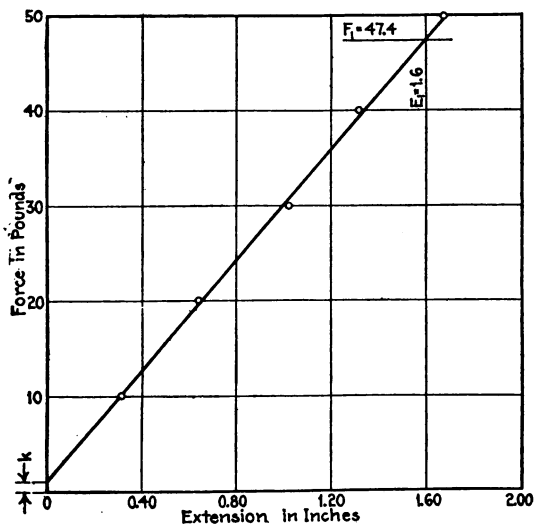


FIG. 5.

(b) **Spring Scale by Method of Least Squares.** The theory of errors shows that the most probable straight line to fit such data is one so located that the sum of the squares of the distances of the plotted points from the line shall be a minimum. This can be calculated as follows. The equation of the line may be written

$$F = SE + k$$

in which  $k$  and  $S$  are unknown, and  $F$  and  $E$  are represented by a number of more or less discordant observations. This equation may be multiplied through by the coefficient of each unknown, thus

$$F = SE + k$$

$$EF = SE^2 + Ek.$$

In each equation the corresponding values of  $E$  and  $F$  are substituted thus forming two series of equations. Each series is summed and the resulting equations are known as the "normal equations" of  $k$  and  $S$  respectively. These may be solved as simultaneous equations to obtain the most probable value of  $S$ .

Using the data of the ascending scale tabulated,

$F = ES + k$	$EF = E^2S + Ek$
10 = 0.32S + k	3.2 = 0.1024S + 0.32k
20 = 0.64S + k	12.8 = 0.4096S + 0.64k
30 = 1.02S + k	30.6 = 1.0404S + 1.02k
40 = 1.32S + k	52.8 = 1.7424S + 1.32k
50 = 1.68S + k	84.0 = 2.8224S + 1.68k
<hr/>	
150 = 4.98S + 5k	183.4 = 6.1172S + 4.98k
Normal equation of $k$ .	Normal equation of $S$ .

From these, by eliminating  $k$ , it is found that  $S = 29.38$  lbs. per inch.

The descending scale is found similarly and the combined scale is obtained by adding the corresponding normal equations from the ascending and descending values.

Note that the numerical values in the equations should be figured to four or five significant figures since, when the equations are solved, the first two or three figures disappear by subtraction. Labor can be saved by using multiples of ten, or simple figures, for the independent variable  $F$ , and by using a pocket-book table of squares to obtain the values of  $E^2$ .

**Problem 21.** Calculate the descending and combined scales for the example given by methods (a) and (b). Compare them.

**Problem 2.** Prove from the fact that the area under each curve equals work done that the descending curve cannot be a straight line.

**Problem 2.** From the ascending data of the example given, figure and compare the different results for the ascending spring scale obtained by the following (faulty) methods.  $50 \div 1.68$ . The average of all the  $F$ 's  $\div$  average of  $E$ 's. The average of  $F \div E$ . The average of each increment of  $F \div$  corresponding increment of extension. The average increment of  $F$  divided by average increment of  $E$ .

## PRESSURE

The measurement of pressure is a special case of force measurement, wherein the area over which the force is distributed is taken into account. Pressure is a compound unit being the number of units of force per unit area.

In calibrating pressure measuring devices, the standard force is that of gravity acting on some standard mass.

Pressure is usually expressed in pounds per square inch. Its measurement is always relative, that is, based upon some other pressure as a datum. A pressure gage reads so many units *above* atmospheric pressure, for instance, and a vacuum gage, so many *below*. Absolute pressure, or pressure counted from zero, is an abstract conception and cannot be measured directly by instruments.

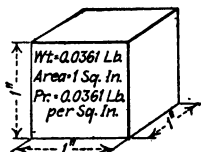


FIG. 6.

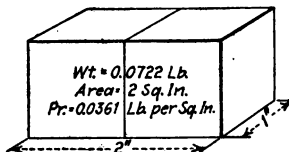


FIG. 7.

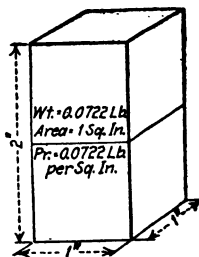


FIG. 8.

Pressure Produced by Water.

Another measure of pressure is the height of water or other liquid which by its weight balances the pressure to be measured.



Fig. 6 represents a cubic inch of water. At 60° F., this weighs 0.0361 lb. Since the area this force acts upon is one square inch, the pressure produced is 0.0361 lb. per square inch. Two cubic inches of water, arranged as in Fig. 7, would have twice the weight, but it would be imposed upon twice the area; hence the pressure would be the same. Arranged as in Fig. 8, however, the area would be one square inch, and therefore the pressure would be twice that shown by Fig. 6. It follows that the pressure produced by a given mass of water varies directly with its height and is independent of its cross-section. Hence, inches or feet of water may be regarded as a unit of pressure.

Fig. 9 shows the application of this principle in the "manometer." The pressure in the chamber *C* is said to be "*X* inches of water" and this equals  $0.0361 \times X$  lbs. per square inch. The absolute pressure in *C* equals this quantity plus the pressure of the atmosphere represented by the arrow at *A*.

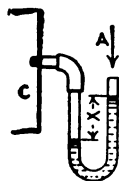


FIG. 9.  
Manometer.

Mercury is also used in manometers, and since its specific gravity is about 13.6 at 60° F., the equivalent is

$$1 \text{ in. of mercury} = 0.0361 \times 13.6 = 0.49 \text{ lbs. per square inch.}$$

### 3. CALIBRATION OF A BOURDON GAGE

**Principles.** The Bourdon gage consists essentially of a hollow circular spring which is deformed when subjected to internal fluid pressure, the deformation causing a pointer to rotate upon a graduated dial. The pointer is readily removed so that it can be set at any part of the dial to correspond to an applied known pressure.

The testing apparatus consists of a chamber in which any desired pressure may be obtained and to which is attached some

accurate device for measuring it. The pressure is generally obtained by some simple form of hand pump. The measuring device is either a "test gage," a column of mercury, or a set of weights acting on a plunger of known area arranged to produce the pressure desired. The test gage is not a desirable standard as it needs calibrating itself from time to time. The mercury

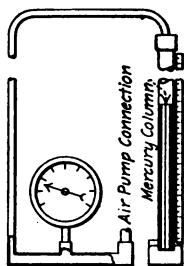


FIG. 10.—Vacuum Gage Tester.

column is a cumbersome and expensive apparatus for pressures above a few pounds, although it is the most accurate method of measuring the true pressure. Besides, a high degree of accuracy is not needed since the least count of commercial gages is generally not less than 5 lbs. It is necessary, however, to use the mercury column when testing vacuum gages, and convenient since they are graduated in inches of mercury and not in pounds per square inch. (See Fig. 10.) The standard weight method is convenient but the friction of

the plunger prevents true records. Fig. 11 shows an apparatus of this type.

(a) **The Constants of the Standard Weight Apparatus** are the area of the plunger and the actual weights of the test weights, from which may be figured the actual pressures produced by them. The area is measured by calipering the plunger. The weights should be determined by comparison with standard weights with a degree of precision compatible with the least count of the gage to be tested. Note that the pressure produced by the plunger and attached pan is always applied.

(b) **Calibration.** Set the pointer to read accurately at the division most used and then take a series of readings of true pressures and instrument readings increasing and decreasing, from which plot a calibration curve. Convenient procedure is as follows. With the weights applied to produce the desired

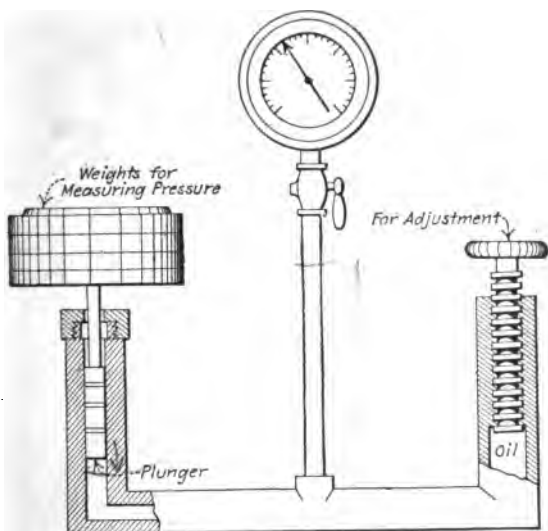
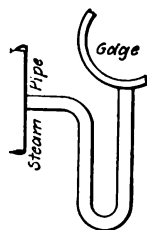


FIG. 11.—Dead Weight Gage Tester.

pressure, depress the plunger a trifle by hand and close the cock between the gage and the pressure chamber, thus confining in the gage a pressure slightly greater than produced by the weights. The pan and weights are now revolved to reduce friction at the plunger and the cock slowly opened. The pressure indicated by the gage will then slowly decrease and a reading may be taken. For increasing values, the pan is raised a trifle, otherwise the procedure is the same. The two readings thus found are added and divided by two to eliminate the effect of friction. If the gage is to be used for ascending pressures, then the calibration resulting from them only should be used. Generally this is not the case and the calibration from the combined readings is preferable.

FIG. 12.  
Gooseneck.

**Problem 3<sub>1</sub>.** Will a gage read lower or higher after continued use? Why?

**Problem 3<sub>2</sub>.** Examine a Bourdon gage to find if there is any device by which a spring too stiff or too weak for the dial markings can be adjusted to them.

**Problem 3<sub>3</sub>.** Will an overheated spring cause high or low indications? Why?

**Problem 3<sub>4</sub>.** Will friction at the plunger of the test apparatus cause plus or minus error in the calibration? Why?

**Problem 3<sub>5</sub>.** Explain how a gooseneck (see Fig. 12) will prevent a gage used on steam from overheating?

#### 4. DRAFT GAGE CALIBRATION

**Principles.** Draft gages are generally special forms of manometers used to measure the reduced pressure (less than atmosphere) in chimneys, which creates the draft. This is expressed in inches of water and, as it is usually only a few tenths, the ordinary U-tube will not do.

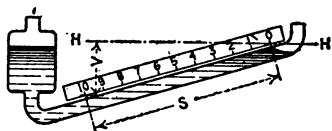


FIG. 13.—Draft Gage.

Fig. 13 shows a favored type. It is the ordinary manometer with one leg on a slant so that the

travel of the level of the liquid used is magnified. In use, the instrument should be set true to its designed level and the liquid adjusted so that the level in the inclined tube stands at zero when there is no difference of pressure.

Calibration of a draft gage can be made by examination of its graduations and the weight of the liquid, or by comparison with a standard instrument.

(a) **Calibration by Calculation.** Having set the gage to its designed level, a horizontal line  $H-H$ , is drawn with a spirit-level through the zero of the scale. The vertical distance  $V$  from the last graduation on the scale to this line is then measured. The level of the liquid in the inclined tube will fall this distance for the whole scale, but the level in the enlarged tube will rise an

amount equal to the length  $S$  of the scale in inches multiplied by the ratio of the squares of the diameters of the bores of the small to the large tube. If water is the liquid, the sum of the rise and fall thus calculated equals the inches of water, pressure. The true pressure corresponding to any other graduation can now be found by proportion.

The calculation of the rise of level in the enlarged tube assumes that the bores are uniform.

Commercial forms of this type of gage generally use oil for the liquid, having a definite specific gravity less than one. When such a gage is calibrated, the total vertical difference of level is found as before. The corresponding height of water is then found by dividing this by the specific gravity of the particular liquid used. The specific gravity may be obtained by using a hydrometer or by weighing a known volume of the liquid.

**(b) Calibration against a Standard Gage.** Connect the gage to be tested with a piece of rubber

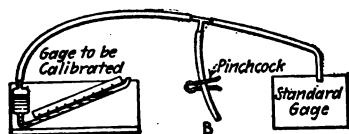


FIG. 14.—Arrangement for Testing.

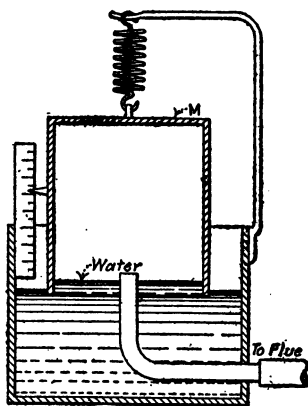


FIG. 15.—Kent's Draft Gage.

tubing to the standard gage as in Fig. 14. The pressure in this tube may readily be reduced to any desired amount by applying the lips to the branch  $B$ , the pinch cock serving to confine the pressure when obtained. Enough readings for a calibration curve should be obtained.

For the standard gage, an impromptu instrument is easily made by using an inclined tube of generous length.

**Problem 4<sub>1</sub>.** If a draft gage using water at 70° F. is correct calculate the rise in temperature that would produce 1 per cent of error. Refer to pocket reference book for densities of water. *Ans., 62°.*

**Problem 4<sub>2</sub>.** In the type of gage shown by Fig. 15, the reduced pressure inside the inverted can *M* causes it to descend against the resistance of the spring (neglecting the buoyancy of the water). If the area of the inverted can is 50 square inches and if the pressure within is  $\frac{1}{16}$  inch of water less than without, what is the total downward force on the can? How much will the spring extend because of this force if the spring scale is 0.2 lb. per inch? *Ans., 0.9 in.*

**Problem 4<sub>3</sub>.** Deduce an equation for the gage shown by Fig. 15, neglecting buoyancy, to show the relation between the extension of the spring and the draft, in inches of water, causing it.

## ANGULAR VELOCITY

The units are usually whole revolutions per minute, abbreviated R.p.m. The simplest form of instrument to measure this quantity is the **hand counter**. This consists of a worm and wheel. The worm is part of a small spindle tipped with rubber so that, if held against the end of a revolving shaft, it turns with the shaft, thus giving a circular motion to the worm wheel. The wheel is graduated so as to indicate the number of turns of the spindle. In operation, the hand counter is applied during a period of time measured with a watch, and from the readings of watch and counter, the revolutions per minute of the shaft are calculated.

**The Continuous Counter**, or cyclometer, is a variation of the hand counter in that its spindle is geared to a series of wheels with numbered faces, partially exposed, so as to show at any time the total number of revolutions. Some forms of continuous counter are driven by a reciprocating lever instead of a revolving spindle. In both forms, the instrument usually receives its motion from a small pin on the end of the shaft whose revolutions are to be measured, which pin acts as a crank.

**The Tachometer** gives instantaneous indications of revolutions per minute without measuring time. The spindle trans-

mitting the speed bears a pair of weights so linked that they move outward from the spindle by centrifugal force, and against the restraint of a spring. In so doing they actuate a pointer on an appropriately graduated dial. As the centrifugal force, and therefore the motion of the weights, are proportional to the speed, the pointer may register the speed instantaneously.

On account of the variation in their internal friction and in the stiffness of their springs with use, tachometers should be calibrated before using on important work.

**The Chronograph** is an instrument by which a graphic record of time is made. Fig. 16 shows a form. Paper is caused to roll over a drum, *D*, by clockwork, and in contact with the paper is a pen on a light arm, *A*, which is caused to vibrate at regular time intervals by connection, electrical or otherwise, with a standard clock. Another arm, *A'*, is similarly actuated by connection with the shaft the speed of which is to be measured, so that it vibrates once each revolution. Thus, there are drawn two lines, one broken at intervals representing time, the other, revolutions; from which may be obtained angular velocity.

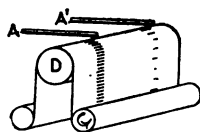


FIG. 16.—Chronograph.

**The Tachagraph** is a very sensitive tachometer, arranged to give an autographic diagram of angular velocity. It is used to measure minute changes of speed within a revolution.

## 5. CALIBRATION OF A TACHOMETER

**Principles.** The standard should be either a continuous counter or a chronograph. In either case it is necessary to have a revolving shaft of variable and controllable speed to cover the whole range of operation of the tachometer. A small steam engine of variable speed may be used, or a water wheel, or a series wound

electric motor. A shunt wound motor will do if a rheostat is put in series with its armature circuit. A small water rheostat is convenient as it gives a wide range of resistance.

**(a) Calibration against a Continuous Counter.** Increasing and decreasing readings of the tachometer should be taken at each speed. The mean of these readings is plotted against the speed as shown by the counter for the calibration curve.

If the continuous counter is arranged on the variable speed shaft so that it can be applied at the same time as the tachometer, the procedure is as follows. Starting with a slightly lower speed than is required, the tachometer is connected and the speed raised to the desired amount. Time and counter readings are then taken with the tachometer still connected. If the speed remains constant as indicated by the tachometer, the observations are valid after the second readings of time and counter are taken. The decreasing readings are obtained similarly.

When the speed is high the counter becomes difficult to read. This may be obviated by an arrangement for throwing the counter in or out of connection with the variable speed shaft.

A hand counter may be used if it can be applied to one end of the shaft while the tachometer is at the other. It is inadvisable to take readings of the two instruments at other than the same time as the speed may vary.

The time during which the revolutions are counted should be of sufficient length to reduce the error of starting and stopping to less than the precision of the tachometer. For instance, if the tachometer cannot be read closer than 2 R.p.m., at any part of its scale, then at 200 R.p.m. the error may be 1 per cent. Now a moving number is unlikely to be timed with an ordinary watch closer than one-half second; therefore the counting should proceed through at least 50 seconds to get precision equal to that of the calibrated instrument. The use of a stop watch will greatly reduce this time as it reduces the starting and stopping error to one-tenth of a second or less.



(b) **Calibration against a Chronograph.** With this apparatus the tachometer may be left in connection with the variable speed shaft, and the speed first increased and then decreased in a series of steps at constant speed. The moving record may be stopped while the speed is being adjusted. The time for counting may be very much reduced, as the chronograph shows the whole number and fraction of turns of the shaft that occur during a time beat which may be as small as desired.

**Problem 5<sub>1</sub>.** Using an ordinary watch, how long should a continuous counter be timed to secure less than 1 per cent of error at 1000 R.p.m.? How long, if a stop watch is used?

*Ans.*, 100 sec.; 20 sec.

**Problem 5<sub>2</sub>.** What is the per cent of probable error in reading a tachometer with a least count of 10 R.p.m. at 1000 and 500 R.p.m.?

*Ans.*, 0.5%; 1%.

## POWER

**Power** is defined as the time rate of work. Quantitatively, work is the product of a force and the distance through which it acts, the unit being foot-pounds. A horse-power is defined as 33,000 foot-pounds of work done in one minute. If an engine can deliver 33,000 foot-pounds of work in one minute it is rated as 1 horse-power; if 66,000, 2 horse-power; and so on.

Generally engines deliver a rotative effort. Suppose, for example, that an engine transmits an average tangential force of  $F$  pounds at its crank pin. (See Fig. 17.)

In one revolution, this force will act through a distance equal to the circle through which the crank pin has passed, or  $2\pi r$ ,  $r$  being the radius of the crank in feet. The foot-pounds of work per revolution are then  $2\pi r \times F$ , and if

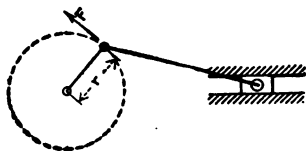


FIG. 17.

there are  $N$  revolutions per minute, the work done per minute will be  $2\pi rF \times N$ , and the horse-power will be

$$\text{H.P.} = \frac{2\pi rFN}{33,000}.$$

The quantity  $rF$  is called the "torque."

**Dynamometers** are used for measuring power. Generally the speed is measured independently of the dynamometer by a hand counter or otherwise, so the dynamometer is applied only to measure the torque. This is done by balancing the torque by a measured force acting at a known distance from the center of rotation.

There are two broad classes of dynamometers: *absorption* by which the power to be measured is converted into some form in which it cannot be used; and *transmission*, by which the power is passed on unchanged.

## 6. CONSTANTS OF FRICTION BRAKES

**Principles.** Friction brakes may be used as absorption dynamometers, and of these the **prony brake** is the commonest and most accurate form. (See Fig. 18.) It consists of a band wrapped around a pulley on the shaft whose power is to be measured, so arranged that by tightening a hand wheel,  $H$ , the friction between the wheel and band can be controlled. The band is held from turning by means of an arm,  $A$ , attached to it, and supported by some force measuring device at its free end. Considering the friction between the band and wheel as a single force,  $f$ , then the horse-power developed is  $2\pi r'fN \div 33,000$ ;  $r'$  being the radius of the wheel in feet. Now since the force  $W$  exerted by the scales,  $S$ , produces equilibrium, by the principle of moments

$$RW = r'f,$$

and therefore by substituting in the expression for horse-power just given

$$\text{H.P.} = \frac{2\pi RWN}{33,000} = BWN.$$

In practice, if the force,  $W$ , and the revolutions per minute,  $N$ , are measured, the horse-power may be calculated, the length of the arm being known.  $B$  is called the "brake constant."

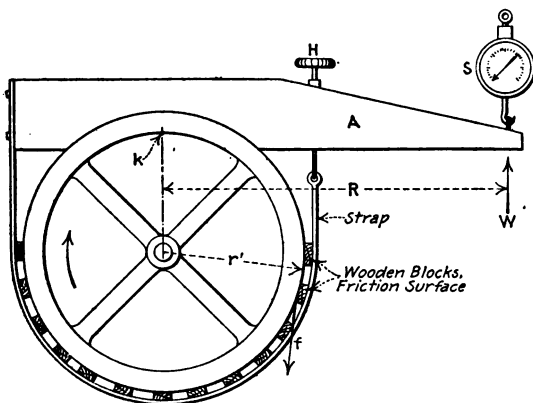


FIG. 18.—Prony Brake.

It should be noted that the arm,  $A$ , by its weight produces a force acting on the scales which should not be included in the force balancing the frictional effort. This should be allowed for by determining the "unbalanced weight" of the brake, or "brake zero," and subtracting it from the scale readings. Methods of determining the brake zero will be given later.

**The rope brake** is a modified form of prony brake. Fig. 19 shows such a one, the friction between the rope and the wheel being balanced by the force of the scales on the right less the weight on the left. The difference between these quantities is

the value of  $W$  in the formula, and the value of  $R$  is the radius of the wheel plus the radius of the rope. There is no unbalanced weight if the lengths of rope on each side of the pulley are equal. The friction, and therefore the horsepower, may be varied by increasing the weight on the right, or by taking more turns of the rope about the wheel; either procedure greatly increases the friction.

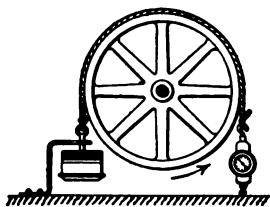


FIG. 19.—Rope Brake.

Fig. 20 gives two other arrangements of rope brake and their equations.

In operation, the heat generated by the friction is removed by circulating water within the brake wheel, the latter being provided with internal flanges for that purpose. It is generally

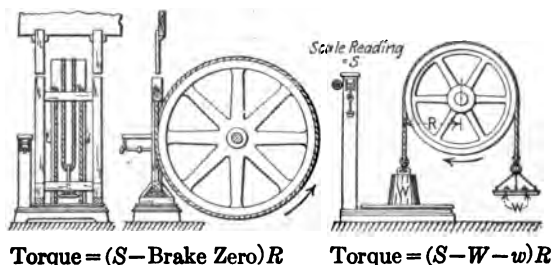


FIG. 20.—Rope Brakes.

sufficient to feed the water at a rate just equal to the loss by steaming.

**Hydraulic friction brakes** are the same as those just described in principle though quite different in detail. A series of discs mounted on the power shaft revolve in a casing filled with water under pressure, the casing being free to revolve about the shaft. The friction between the discs and the water causes a turning

effort upon the casing which is balanced by a force measuring device the same as with the common prony brake. Regulation is secured by, varying the water pressure. The constants are the same as for the other types, but in most cases there is no unbalanced weight.

(a) **Determination of Unbalanced Weight.** If friction could be entirely eliminated between the band and flywheel, the unbalanced weight would be the reading of the scale, Fig. 18. But, no matter how loose the band is, there is always some friction between it and the wheel tending to hold up the brake arm when the wheel is stationary. If the brake is removed from the wheel and supported by a knife edge or circular pin at the point *k*, Fig. 18, the other end being supported by the scales as in operation, then the scales will indicate the unbalanced weight provided that the flexible band does not change in its weight distribution. This is one method of finding this quantity.

Another method consists in revolving the brake wheel first in one direction and then in the other, and noting the corresponding readings of the brake scales. With the wheel revolving clockwise, Fig. 18, the scales will indicate the unbalanced weight *plus* the force necessary to balance the friction at the band. Anti-clockwise, it indicates the unbalanced weight *minus* this force, since the friction is reversed. Calling the force balancing friction *X*,

$$\text{Unbalanced Wt.} + X = \text{1st reading}$$

$$\text{Unbalanced Wt.} - X = \text{2d reading}$$

$$\text{Adding, } \text{Unbalanced Wt.} = (\text{1st} + \text{2d reading}) \div 2.$$

When applying this method it is necessary that *X* remain constant. The band should be quite loose, the bearing blocks wet with oil, and the wheel turned at a uniform rate.

The third method is the same in principle as the second, but the brake is revolved instead of the wheel. The spring balance, Fig. 18, is first drawn upward giving the weight  $+X$  reading; then the weight of the brake is allowed to draw it

down for the weight  $-X$  reading. Note that if the spring balance is slanted out of its correct position relative to the brake, a component force will be indicated.

(b) **Determination of the Horse-power per Pound of Thrust per Revolution.** This is the "brake constant" or  $2\pi R \div 33,000$ .  $R$  should be measured with a tape or measuring rod. Note that  $R$  is the *perpendicular* distance from the center of the brake-wheel to the line of the balancing force  $W$ , Fig. 18.

**Problem 6<sub>1</sub>.** What is the break constant if  $R=5$  ft.,  $3\frac{1}{2}$  in.? If unbalanced weight of the brake is 2.5 lbs., what scale reading would be necessary to balance 40 H.P. at 160 R.p.m.?  
*Ans.*, .001; 252.5 lbs.

**Problem 6<sub>2</sub>.** By the third method, the scales indicate 10 lbs. when the brake is pulled up. (Fig. 18.) Its weight is not enough for it to drop, consequently the balance is reversed and the brake *pulled* down, for the weight  $-X$  reading. The scales then indicate 2 lbs. What is the unbalanced weight?  
*Ans.*, 4 lbs.

**Problem 6<sub>3</sub>.** If the unbalanced weight is 5 lbs., where should weight be added to balance the brake, and how much?

**Problem 6<sub>4</sub>.** Given an arrangement like Fig. 18 except that the wheel turns anti-clockwise and the spring balance is inverted. If the R.p.m.=100, arm=4 ft., unbalanced weight=14 lbs., and the balance reads 20 lbs., what is the horse-power?  
*Ans.*, 2.59 H.P.

## 7. CALIBRATION OF A FAN BRAKE

**Principles.** Fan brakes are convenient for measuring the output of high speed motors that operate at variable speed, such as automobile engines. Fig. 21 shows such a one. The energy of the shaft is absorbed by imparting kinetic energy and heat to the air. With a fixed set of vanes the power thus absorbed varies as the cube of the speed. The capacity of the brake at a given speed of rotation can be changed in only two ways, namely, by changing the size of the vanes or their distance from the center of the shaft. The first method may be taken by swinging the vanes around on their arms so that the effective area resist-

ing motion is lessened. For the second method, the fastenings of the vanes to the arms may permit the desired radial variation.

This type of dynamometer is sometimes mounted on a frame independent of the engine shaft and driven by a belt. In this case, the belt losses and brake journal friction (more or less indeterminate quantities) are added to the resisting effort.

(a) **Calibration against a Transmission Dynamometer.** By means of a transmission dynamometer, simultaneous values of torque necessary to drive the fan and revolutions per minute are determined for a given adjustment of the vanes. From these values, the horse-power absorbed at each speed may be calculated, and a curve of horse-power vs. revolutions per minute plotted. This curve may be used for finding the horse-power, when the brake is in usual operation, from readings of the rotative speed.

(b) **Calibration against a Calibrated Motor.** If the fan brake is belt driven, it may be run by a variable speed electric motor, through the same belt and pulleys to be used when the fan measures power. At a definite speed of the fan, readings of the armature and field currents of the motor and of its speed are taken. The belt is then removed, a prony brake applied to the motor pulley, and the horse-power of the motor measured under the same conditions of current and speed. If all controllable conditions are the same, this horse-power will be the same as that absorbed by the fan at the applied speed. A series of similar trials at different speeds will give data for a calibration curve.

This calibration will be somewhat in error owing to the fact that the reaction on the motor bearing caused by the prony brake thrust is somewhat different from the reaction due to the belt pull; hence the friction of the motor under the two loads will be slightly different.

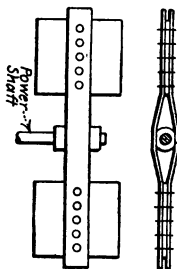


FIG. 21.—Fan Brake.

(c) **Use of the Horse-power Constant.** The power absorbed by the fan for a given setting of the vanes equals

$$K \times \text{density of the air} \times (\text{R.p.m.})^3$$

in which  $K$  is a quantity very nearly constant. Assuming that the density of the air also is nearly constant, the product of  $K$  and the density may be found experimentally at a convenient speed. Then the horse-power at any other speed may be found, approximately, by multiplying this product by the cube of the speed.

The experimental determination of  $K$  times air density is made by taking readings according to either method (a) or (b). Then the desired value,  $K$ , equals the horse-power divided by the cube of the applied speed.

Variations in belt losses and in the value of  $K$  make this method inexact.

**Problem 7.** What percentage of error will be caused by the density of the air changing from that corresponding to a barometer of 29.5 in. of mercury, at which the fan is calibrated, to 30 in.? What percentage by a change of temperature from 60 to 80° F.?

*Ans.*, 1.7%; 3.8%.

## 8. CALIBRATION OF A TRANSMISSION DYNAMOMETER. WEIGHT-ARM TYPE

**Principles.** The weight-arm type of transmission dynamometer consists of some device by which the torque of a revolving shaft can be balanced by a standard weight or weights acting with a known leverage. The balancing torque caused by the weights is thus a measure of the horse-power when the revolutions per minute are known. There are many forms of this type of dynamometer differing mainly in mechanical details.

**Belt dynamometers** of different kinds are in the weight-arm class, of which Fig. 22 represents one. The torque of the trans-



mission shaft is equal to the effective belt pull,  $T_1 - T_2$ , multiplied by the radius at which it acts,  $r$ . Disregarding the friction at the bearings of the pulleys,  $p$  and  $p'$ , the reactions at these bearings will be  $2T_1$  and  $2T_2$ , respectively, (as will be seen from a consideration of the belt forces.) Taking moments about the fulcrum  $f$ , and disregarding the weight of the arm  $A$ , we have

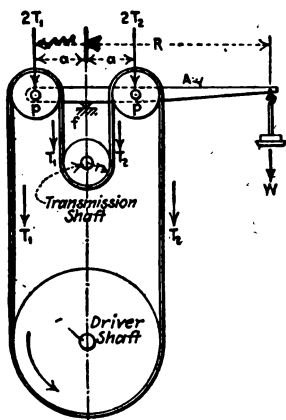


FIG. 22.—Belt Dynamometer.

$$WR + 2T_2a = 2T_1a$$

from which

$$T_1 - T_2 = WR \div 2a$$

and the torque

$$= r(T_1 - T_2) = rWR \div 2a.$$

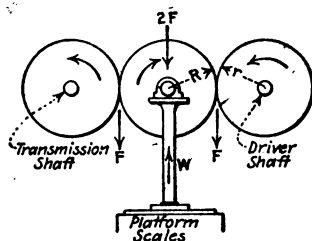


FIG. 23.—Pillow Block Dynamometer.

Thus by balancing the belt torque by a known weight  $W$ , the power may be measured if  $R/a$  is known and if a suitable allowance is made for friction.

A pillow block dynamometer is shown by Fig. 23, consisting of three gear wheels the middle one of which is mounted on a pillow block, so as to bear freely on a weighing device. The turning force of the driver,  $F$ , produces an equal resisting force at the gear on the transmission shaft, barring friction. The reaction  $W$ , which can be weighed by the scales, thus equals  $2F$  and hence the torque of the transmission shaft is

$$Fr = \frac{W}{2} \times r.$$

The reaction  $W$  may also be measured by hanging the middle wheel from one arm of a lever above the other carrying a balancing weight.

Some forms of pulley block dynamometer omit the middle wheel and weigh the reaction of the bearing on the transmission shaft directly. When this is done, the transmission shaft must be arranged to rest freely on the weighing device.

The Webber differential dynamometer is a variation of Fig. 23, but similar in principle. It is shown diagrammatically by Fig. 24, which represents three bevel gears, the middle one being

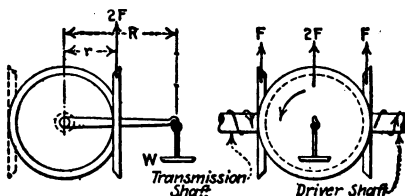


FIG. 24.—Webber Dynamometer.

supported by a lever whose fulcrum lies on the common axis of the driver and transmission shafts. Similar to the principle shown by Fig. 23 the net force on the middle gear is  $2F$ . This is balanced by the dead weight  $W$  acting with a lever arm of  $R$  feet. Taking moments of the forces acting on the lever around its fulcrum, we have

$$2Fr = WR$$

from which, torque transmitted =

$$Fr = \frac{W}{2}R.$$

As actually constructed there is another gear on the lever-arm, as indicated by the dotted line. This does not change the relation, merely substituting for the moment  $2Fr$  two moments,

$Fr + Fr$ . Also there is a jockey weight on the arm in addition to the pan weights similar to the arrangement of a platform scales.

**The Emerson power scale** is a dynamometer by which the torque is passed to a transmission pulley on the driver shaft through a lever system so arranged that a thrust in an axial direction is given to a non-revolving sleeve on the shaft. This thrust is balanced by dead weights which consequently measure the torque.

**Friction** of the moving parts of the dynamometer itself has not been accounted for in the equations. Generally the dynamometer indicates the torque necessary to overcome its own friction and windage in addition to the external torque on the transmission pulley which alone it is the purpose to measure.

**(a) The Constants of a Dynamometer of the Weight-arm Type** are first, the true values of the dynamometer weights; second, the unbalanced weight of the arm or lever system; and third, the leverage ratio produced by the arm or lever system.

**The true values of the weights** should be determined with a scales sufficiently precise to keep within a reasonable percentage of error.

**The unbalanced weight of the arm** may be found as for a prony brake, by revolving the driver shaft in first one direction and then the other by hand, and noting the resulting force at the end of the arm where the weights are to be applied. A spring balance may be used for this purpose. Half the sum of the two readings equals the unbalanced weight.

**The ratio of levers** may be obtained generally by direct measurement. In the case of the Webber dynamometer, for instance, only the length of  $R$ , Fig. 24, is necessary. For the arrangement of Fig. 22, the ratio is  $R a$ , as shown by the equation.

**(b) Calibration by Calculation.** The torque equivalent to each dynamometer weight acting at the previously determined leverage is found by multiplying its value in pounds by the lever-

age ratio as shown in the equation for the dynamometer in question. A series of such determinations furnishes data by which the horse-power corresponding to any weight may be found when the dynamometer is in usual operation.

The torque equivalent to the unbalanced weight of the arm should be added to that indicated by the weights applied if the turning effort of the driver on the arm is upward. If it is downward, the power being then gaged by scales instead of weights simply, the torque equivalent to the unbalanced weight of the arm should be subtracted.

**(c) Allowance for Friction, Windage, and Centrifugal Force.**

To allow for friction, a crude but convenient approximation is based upon the assumption that the frictional resistance is the same under all conditions of external torque. If, then, the dynamometer is run with the transmission shaft entirely free, the weight to balance the arm gives the correction to be subtracted from the readings in usual operation. Windage is included in this, but if the dynamometer is to be used at various speeds, similar corrections should be determined at these speeds to allow for the varying value of the windage.

With some dynamometers having revolving levers, bell-cranks, and the like, such as the Emerson power scale, centrifugal force acting on these parts may cause a distortion of the indications. This is allowed for under the windage corrections.

**(d) Comparison with Prony Brake Measurements** is the most valid means of calibrating any transmission dynamometer. To do this, a prony brake is fitted to the transmission pulley and is adjusted so as to balance each

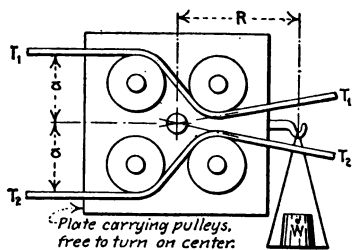


FIG. 25.—Belt Dynamometer.

dynamometer weight in turn, power being delivered as in usual

operation. The torque shown by the prony brake is the true torque measured by the corresponding weight. When a jockey weight is used as for the Webber dynamometer, a calibration curve of torque against jockey weight positions is convenient.

If the dynamometer is to be used at various speeds, it should be calibrated at a number of them, the results to be in terms of either horse-power or torque, as shown by the brake, corresponding to each dynamometer weight. In this way friction, windage, and centrifugal force may be taken into account.

**Problem 8.** Deduce the torque equation for Fig. 25.

## 9. CALIBRATION OF A TRANSMISSION DYNAMOMETER, SPRING TYPE

**Principles.** Spring dynamometers differ from the weight-arm type in that the torque to be measured is balanced by a spring or springs through which the torque is passed. The spring is consequently deformed by either a tensile, compressive, or twisting stress. If the constant of the spring is known (that is, the number of pound-feet of torque necessary to cause a unit deformation) then by noting the deformation, the horse-power may be determined.

Fig. 26 shows, in part, the principle of the **Van Winkle dynamometer**. Power is taken off at the loose pulley *P* which is driven through springs by the disc attached to the driver shaft. The resulting deformation of the springs permits a change of position of the pulley relative to the disc, and this operates a bell-crank lever (not shown) which in turn actuates a pointer on a stationary dial. The dial is arranged to indicate horse-power direct.

Another spring dynamometer, made by the **Central Laboratory Supply Company**, is shown by Fig. 27. Two shafts are connected by a spring through which the power to be measured is passed. These shafts are provided with discs arranged as commutators,

being insulated from the shafts except at the shaded portions shown under the brushes. In this position, an electric circuit is

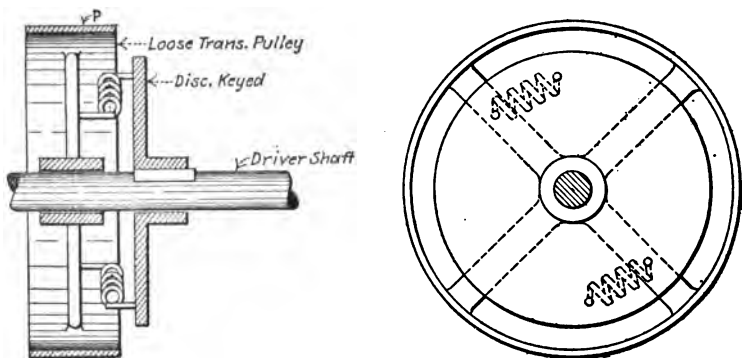


FIG. 26.—Van Winkle Dynamometer.

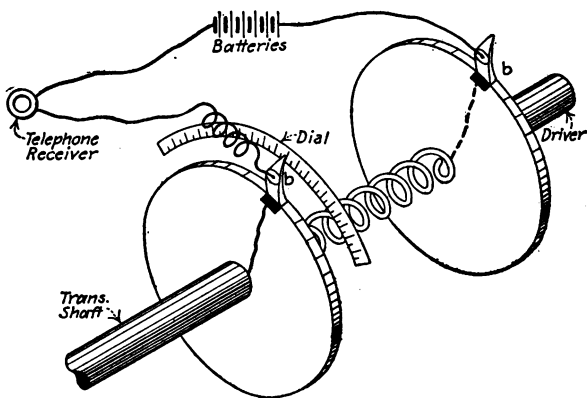


FIG. 27.—Central Laboratory Dynamometer.

completed causing a click in a telephone receiver. The right-hand brush is stationary, but the one on the left is so arranged that it may be swung around the shaft. In operation, the latter is

manipulated until a click is heard in the receiver, indicating that both contact pieces are passing under the brushes at the same time. Then the angular motion of the left-hand brush shown on the dial and measured from its clicking position when there is no torque delivered, is equal to the twist of the spring, and hence is a measure of the torque.

In usual operation small variations of the torque, due to belt flapping, etc., make the clicking position somewhat variable. It is therefore convenient to read the maximum and minimum angles at which no click is heard and to take the average of these as the twist of the spring.

**The Flather dynamometer** (Fig. 28) differs from the foregoing in that the torque is communicated to the transmission pulley through small pistons, *p*, working in cylinders filled with oil.

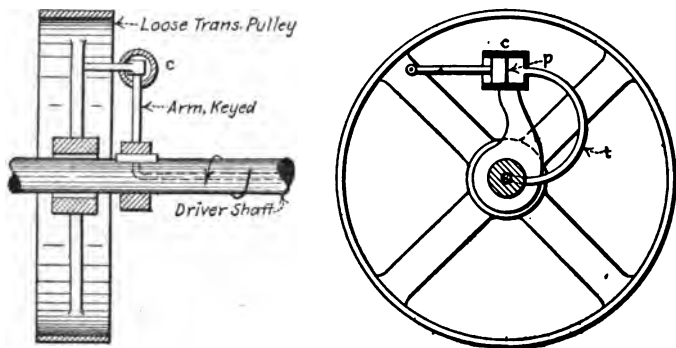


FIG. 28.—Flather Dynamometer.

The fluid pressure thus produced is transmitted through tubes, *t*, and a longitudinal hole through the center of the driver shaft. Connecting with this hole at the end of the shaft is an indicator by which a graphic record of the pressure is made. Since this pressure is proportional to the torque (being produced by the driving force at the cylinders, *c*, acting at a constant distance from the shaft center) it is a measure of the horse-power.

**Friction** affects the indications of a spring dynamometer in two ways. First, since the friction of the loose transmission pulley or of the transmission shaft on its bearings always acts *against* the motion, it makes the indicated torque *greater* than the true external torque. Second, friction of the indicating mechanism makes the indications for increasing torques low, and for decreasing high, similar to the action of friction in a pressure gage. Generally the first effect is greater than the second.

(a) **Static Calibration.** With the driver shaft clamped, the transmitting shaft may be twisted with a series of known weights acting at a measured distance from the center of the shaft. Thus a series of values of torque may be obtained with the corresponding instrument indications, the latter being either in dial graduations, degrees of twist, or height of an autographic diagram, as may be appropriate to the instrument tested. To apply the static torque, the transmission pulley may be used as the arm, a rope being tied to a spoke and passed over the pulley face from which to hang the weights. If the necessary number or size of standard weights are not available, a lever may be clamped to the transmission shaft, and a single weight applied at various distances from the center. In this case, the moment of the lever should be accounted for.

Increasing and decreasing values of the torque should be applied and corresponding readings taken to eliminate the effect of friction. For the increasing readings, the weights are caused to bring up the torque gradually to the desired amount. For decreasing, extra torque is brought upon the shaft by bearing on the weights by hand; then gradually removing the hand pressure so that the torque will decrease to the desired value. The average of each pair of readings is then plotted against torque for a calibration curve. By this procedure, for increasing values, the motion of the transmission pulley or shaft is opposite to that for decreasing values. Hence both effects of friction, previously noted, are eliminated.



For dynamometers with which the angle of torsion is read, the curve of torque vs. degrees may be used to get the spring constant. Then if  $S$  is this constant in pound-feet per degree of twist and  $A$  the torsion angle noted in usual operation,

$$\text{horse-power} = .00019 \times S \times A \times \text{R.p.m.}$$

which is the horse-power delivered by the dynamometer pulley.

**(b) Allowance for Friction, Windage, and Centrifugal Force.**

The readings of the dynamometer should be reduced by an amount corresponding to the torque necessary to overcome these forces. The values of the corrections may be determined as for weight-arm dynamometers. For recording instruments, a line should first be made on the chart with the dynamometer running free. The diagram under load should then be measured from this friction line as a datum.

For dynamometers using a measurement of the torsion angle, the correction is made in degrees, being subtracted from the reading observed in operation.

**(c) Comparison with a Prony Brake** may be made in exactly the same way as for weight-arm dynamometers. (Test 8, c.)

**Problem 9<sub>1</sub>.** The reading of a spring dynamometer running at 450 R.p.m. is 26 degrees. If the spring constant is 3 lb.-ins. per degree, and if the correction for windage, friction, etc., is 5 degrees, how many foot-pounds of work will be done in ninety seconds? What will be the horse-power transmitted?

*Ans.* 3540 ft.-lbs.; .0715 H.P.

**Problem 9<sub>2</sub>.** Same data as Problem 9<sub>1</sub>, except that the friction of the transmission shaft is separately determined, the correction being 4 degrees, and the correction for friction of the indicating device is 1 degree. What is the transmitted horse-power when the torque is increasing? When it is decreasing?

**Problem 9<sub>3</sub>.** The axis of the piston  $p$  (Fig. 28) is 15 inches from the center of the driver shaft, and the diameter of the piston is 2 ins. Disregarding friction, what fluid pressure is produced when the horse-power is five, at 150 R.p.m.?

*Ans.* 44.6 lbs. per sq. in.

## THE ENGINE INDICATOR—REDUCING MOTIONS

The engine indicator is an instrument which makes a graphic diagram giving the relation between the pressure and volume of the fluid in an engine cylinder under working conditions. Since the area of such a diagram is proportional to work, the

indicator is a dynamometer of a special type. It is shown in principle by Fig. 29. *C* is a small cylinder with a close fitting piston *P* which is subject to the same fluid pressure as in the engine cylinder, being connected to it by a short

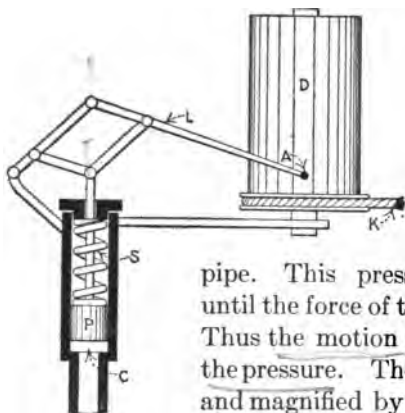


FIG. 29.—Engine Indicator.

pipe. This pressure compresses the spring *S* until the force of the spring balances the pressure. Thus the motion of the piston is proportional to the pressure. The piston motion is communicated and magnified by means of a linkage *L* bearing a pencil point at *A*. *D* is a metal drum free to oscillate on a spindle and carrying the paper on which the record is to be made. The drum is

actuated by the engine cross-head through a cord *K*, a spring within the drum serving to bring it back upon the return stroke. Thus the motion of the drum is proportional to the engine piston and therefore to the volume in the engine cylinder behind the piston. The diagram made by the pencil point on the record paper is one of pressure shown vertically and volume (or piston stroke) horizontally.

The principal use to which the indicator is put is the finding of the *mean effective pressure* in an engine cylinder throughout its working stroke, from which quantity the *cylinder or*

indicated horse-power may be calculated. (See Test 41.) Fig. 30 is a typical indicator diagram from a steam engine. The average pressure on the forward stroke is the mean height, to scale, of the curve *abc*. On the return stroke, the average pressure (back pressure) is the mean height of *cde*. The effective pressure is the difference between these two, or the mean height of the indicator diagram. This may be found by dividing its area in square inches by its length in inches, and multiplying the quotient by the scale of pressure. The scale of pressure is

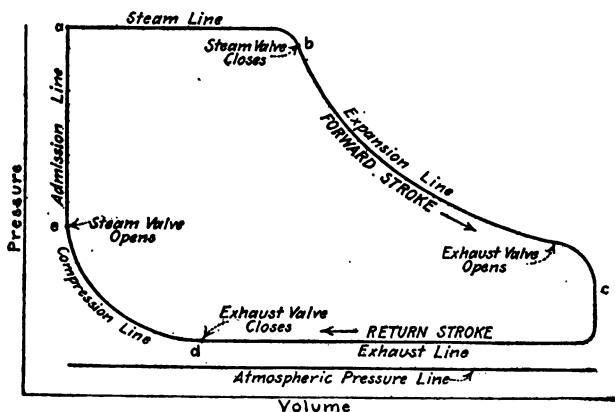


FIG. 30.—Indicator Diagram from Steam Engine.

the number of pounds per square inch on the indicator piston necessary to produce one inch of rise of the pencil, and is referred to as the *spring scale*.

The indicator is equipped with a number of springs of different stiffnesses so that one appropriate to given conditions may be selected.

The drum spring is adjustable so that a greater tension may be operative at higher rotative speeds to provide proper acceleration of the drum upon the return stroke.

The cord actuating the drum does not receive its motion directly from the engine cross-head, but from a *reducing motion*, the function of which is to reproduce the engine stroke on a small, but always proportionate scale. A pantagraph is often used for this purpose. Under Tests 12 and 13 are described various types.

## 10. CALIBRATION OF THE INDICATOR SPRING AND PENCIL MOTION

**Principles.** There are four causes of error in the ordinates of an indicator diagram. First, when applied to high speed engines, the inertia of the indicator piston and attached linkage causes a variation from correct positions. Second, at high speed, the pressure in the indicator cylinder lags behind that operating on the engine piston, because of the inability of the steam immediately to traverse the passages to the indicator. Third, the mechanism actuating the pencil may incorrectly magnify the motion of the indicator piston. Fourth, the spring scale may not be exactly known. Friction of the indicator piston and linkage causes a variation of the spring scale, since because of it the pencil is too low when rising and too high when falling. (See Test 2, principles.)

The first of these errors may be avoided, or reduced, by the use of stiffer springs than are appropriate to low rotative speeds. With a stiffer spring, the total rise of the pencil is less, and therefore the velocity and inertia of the pencil motion parts are decreased.

The second cause of error, lag in the fluid pressure, may be reduced by the use of short and direct pipe connections between the indicator and engine cylinders.

Errors due to the third and fourth causes may be corrected as will be described.

(a) **The Pencil Motion may be Tested as Follows:** With the indicator spring removed, a horizontal line is drawn on a piece of record paper placed on the drum, by revolving it by hand, the pencil bearing against the paper at a low position of the linkage. Then, with the drum held stationary, a vertical line is made by moving the pencil and linkage up by hand. This is repeated with the drum in a second position. The two vertical lines should be parallel and straight, and perpendicular to the horizontal line, if the pencil motion is true.

(b) **Determination of Ascending, Descending, and Combined Spring Scales by Graphic Method.** It is necessary to get a series of values of true pressures and corresponding heights of indicator pencil. The apparatus for varying and measuring the pressure should preferably be one using the same working fluid to which the indicator is subjected in practice, so as to duplicate the conditions of temperature and friction. Dead weights applied directly to the indicator piston are sometimes used, but these do not correctly reproduce the working conditions. Fig. 31 represents a calibration apparatus using steam. The pressure is varied by manipulating valves *A* and *B*, an opening of *A* and closing of *B* having the effect of increasing the pressure in the large steam chamber, and vice versa. The measuring device is a set of known weights acting on a plunger of known area, from which the pressure balancing the weights may be figured. An accurate and precise Bourdon gage would serve the purpose as well. The gage shown in Fig. 31 is used to indicate the pressure in the chamber when the weights are not balanced, for convenience in manipulating the valves *A* and *B*.

Using the calibration apparatus, a diagram similar to Fig. 32 is made on an indicator card. For the ascending pressures, the weight table must be balanced and rising very slowly when the line is drawn. Similarly, for descending pressures, the table should be gently falling. The table is revolved by hand to reduce friction at the plunger.

The heights of the lines from the atmosphere line are then measured to  $\frac{1}{16}$  in., and recorded on the diagram with corresponding pressures as in Fig. 32. These data are then plotted and results obtained as for Test 2 (a).

(c) **Spring Scales by Method of Least Squares.** The same experimental data are used, but the results are calculated as for Test 2 (b), the value of  $F$  in the equations then being the observed

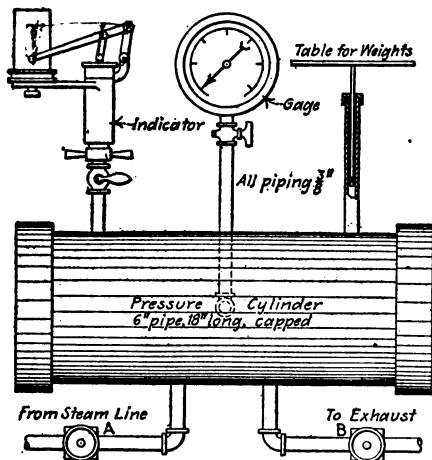


FIG. 31.—Indicator Spring Testing Apparatus.

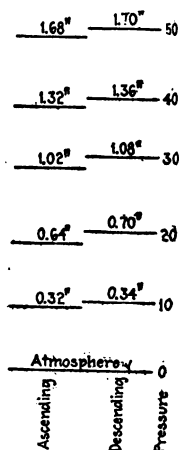


FIG. 32.—Calibration Records.

pressure in pounds per square inch, and  $E$ , the height of the indicator pencil in inches.

(d) **Correction of Indicator Diagrams.** Various methods, more or less accurate and laborious, have been proposed for applying calibration results to indicator diagrams. Generally, it is sufficient to use the combined spring scale with which to multiply the mean height to get the mean effective pressure, the error involved being within the limit of accuracy of power tests. But the combined spring scale represents the true scale of the spring

more nearly than it does actual conditions, since the method of figuring it eliminates friction. Strictly speaking, the ascending and descending scales should be used separately on the diagram, the former applied to the mean height of *cde*, Fig. 30, since the pencil is rising on that line; and the latter to the mean height of *abc* since there the pressure falls. But when the back pressure line is horizontal in large part, as it almost always is, the descending scale applied to the whole diagram will yield a fair result.

The indicator is sometimes applied to other than power measurements, for which a high degree of accuracy is desirable.

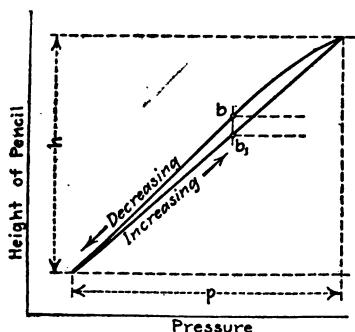


FIG. 33.

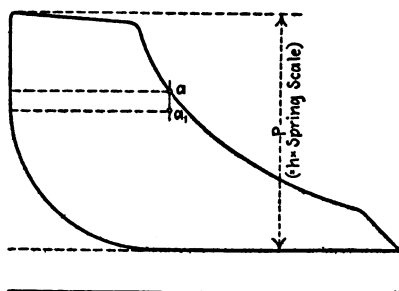


FIG. 34.

### Correction of Indicator Diagrams.

It is then necessary to reconstruct the diagram to get correct results. A method of doing this is as follows. Calibration data are obtained as previously described, and plotted as shown by Fig. 33. The range of pressure  $P$  for the calibration must be the same as that in the indicator diagrams to be corrected, and all operating conditions of the indicator during its test the same as when the diagrams are taken. It will be noticed that the descending curve of Fig. 33 is not straight, and is joined to the ascending curve. Careful experimentation will reveal these characteristics. Actually, there can be no unfilled gap between

the two curves, as there would be under the assumption that both are straight lines, an assumption, as was pointed out under Test 2, made only for convenience in approximately figuring the scales.

Having plotted the curves according to Fig. 33, the *ascending* scale should then be obtained by either of the two methods (a) or (b) previously given. The calibration curve is next laid alongside of the indicator diagram to be corrected as shown by Figs. 33 and 34. Now, the back pressure line is shown correctly to the ascending scale determined from the calibration. To represent a point *a* of the indicator diagram on this scale, the construction  $abb_1a_1$ , is used, the point  $a_1$ , being the required corrected position of *a*. Enough points are corrected in this way to reconstruct the pressure line of the indicator diagram. The ascending scale will then apply correctly to the whole diagram.

(e) **Sampling.** When there are a large number of diagrams for a single engine test, a few representative ones only are selected for reconstruction. Suppose, for instance, that there are 24 diagrams and that the average of all their mean heights is *H* inches. Three of these should be selected as near as possible to this average mean height, and fairly representative in general proportions. These three diagrams are reconstructed and then the desired results obtained from them.

To get the average of the mean effective pressure from all the diagrams without reconstructing all of them, the value *H* may be reduced in the proportion that the mean height of the three sample diagrams is reduced after reconstruction. The corrected value of *H* when multiplied by the ascending spring scale gives the corrected mean effective pressure.

**Problem 10<sub>1</sub>.** The pencil of an indicator throws 2 ins. If the boiler pressure is 145 lbs., what should the spring scale be to give as high a diagram as possible?

*Ans.*, 80.

**Problem 10<sub>2</sub>.** The ratio of the pencil motion to the piston motion of an indicator is 6 : 1. The area of the piston is  $\frac{1}{2}$  sq. in. With a 30-lb. spring, the difference between ascending and descending positions of the pencil at



a given pressure is 0.04 in. If the friction is all at the piston, how much is it in pounds? *Ans.*, 0.6 lb.

**Problem 10.** With the same indicator and spring as last problem, if the friction between the pencil point and the paper is 2 oz., how much difference in the height of the pencil will this make? *Ans.*, 0.0125 inch.

**Problem 10.** With the apparatus of Fig. 31, what is the effect of friction at the plunger upon the apparent true pressure? What effect has this upon the calibration records?

## 11. TESTING THE MOTION OF THE INDICATOR DRUM\*

**Principles.** When an indicator drum is driven by a cord attached to a reducing motion mechanically correct, the motion imparted to it is approximately simple harmonic. Upon the forward stroke of the engine, the cord is pulling the drum, and upon the return stroke, the drum spring is stressing the cord to keep it taut. Thus the cord is always under stress. If the stress varies, the cord will stretch according to such variation, and consequently the drum will not assume the correct positions it would have if driven by an inelastic connector.

Stress in an indicator cord is the resultant of three distinct forces: namely, the drum spring tension, the force required to overcome the inertia of the drum, and the force to overcome friction at the drum spindle and at any guide pulleys used to guide the cord. The force of the spring increases with the forward stroke of the drum (the spring being wound up) and decreases upon the return. The variation is usually uniform with the motion.

In general, at the beginning of the forward stroke, inertia increases the cord stress since, as the speed is increasing from zero to a maximum at mid-stroke, inertia effects a tendency of the drum to lag. Beyond mid-stroke, however, the speed is decreasing, and as the drum tends to exceed the velocity induced

\*For a more complete discussion of this subject see author's article in *Power*, Aug. 20, 1912.

by the cord, a slackening results. Upon the return stroke the force of acceleration varies in the same way as on the advance.

The force of friction is always opposite to the drum motion and therefore changes its direction at each stroke. During the forward stroke it tends to increase the cord stress, and upon the return, to decrease it.

These three forces are represented graphically by Fig. 35. The resultant stress in the cord equals their algebraic sum at any drum position. At low speeds, the force of acceleration

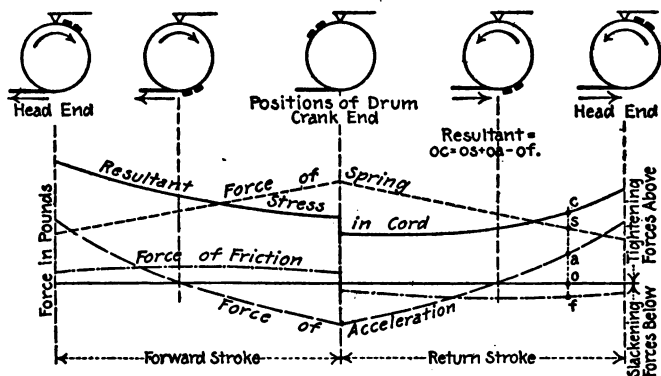


FIG. 35.—Variation of Forces on Indicator Card.

is practically negligible; hence the resultant cord stress increases on the forward stroke, and decreases on the return with lowered values due to the reversal of friction. At high speeds, the force of acceleration reverses this variation as shown by Fig. 35. At some intermediate speed the force of acceleration and the spring tension nearly balance, so that the cord stress is approximately constant during each stroke, but a trifle less on the return because of friction. This is the speed at which the indicator drum is best adapted to work, since the more nearly constant the cord stress is, the less is its stretch and the consequent error.

Let us now compare the effects of a higher speed than this with the ideal condition of constant cord stress. The effect of speed is to increase the stress at the head end and decrease it at the crank end (see Fig. 35). At the head end, therefore, the cord is longer and the drum travels further in this direction. Similarly, at the crank end, the cord is shorter and the drum is pulled further toward that end. The net effect is to pull out the indicator diagram. Now, if the diagram were lengthened *uniformly* its proportions would remain correct, and there would be no error. Accordingly, if the cord were subjected to a uniformly decreasing and increasing stress, and if its stretch varied directly with the stress, a correct diagram would result. It follows that, lacking such uniformity, the error of any point of the diagram should not be judged by the absolute stress or stretch at that point, but by the difference between this stress or stretch and that necessary to produce a uniform lengthening. It is seen from Fig. 35 that the resultant cord stress is greater than that necessary for uniformity on the forward stroke and less on the return. Hence, during the forward stroke, the cord is too long and a point on the indicator diagram is to the left and behind its correct position. During the return stroke the cord is too short; a point on the indicator diagram is to the right and again behind its correct position, since the motion is reversed. The net effect of the stretch of the cord, then, is to make the mean effective pressure appear smaller, and the cut-off, compression and release earlier than their true values.

(a) **Determination of Spring Tension.** Since this varies through the drum stroke, its value at mid-stroke may be measured, or the average of the values at the ends of the stroke may be used. A spring balance of between 5 and 10 lbs. capacity should be fastened to the cord leading from the drum, and through it the drum pulled past the position at which it is desired to measure the spring tension. The reading of the balance is noted at the instant of passing this position. This equals the spring tension

plus friction. The spring is then allowed to reverse the motion, and another reading is taken, equal to the tension minus friction. The sum of these readings divided by two is the spring tension.

**(b) Determination of Drum Friction.** The same method is used as for measuring the spring tension. The friction equals the *difference* of the readings divided by two.

With an indicator properly lubricated and adjusted, and of good make, the friction should not exceed 5 or 10 per cent. When the cord runs over guide pulleys, however, the friction is greatly increased. When the friction is high, the cause should be looked for and remedied.

**(c) Testing Indicator Cord.** Tie one end of about four feet of the cord to be tested to a fixed point on a bench or table, and the other end to a spring balance. Mark this end a few inches from the balance with a fine ink line, and under this line place a piece of paper or a foot-rule. Stretch the cord by pulling the balance horizontally until about 5 lbs. are indicated. Now reduce the force to about 1 lb. and repeat this procedure a few times. The elongation may then be noted for the applied range of stress. As the cord in the operation of the indicator is generally not stressed more than 5 lbs. or less than 1 lb., this range is appropriate. A good cord should not stretch more than 0.01 in. per foot per pound, but grades will be found with four times this stretch and more.

The elongation and contraction of the cord are very much greater at stresses less than 1 lb. On this account there may be marked overtravel at the crank end of the drum motion without a visible slackening or vibrating of the cord when the indicator is in usual operation.

**(d) Adjustment of Drum Spring Tension.** The indicator is put in working adjustment on the engine to be indicated, the piston spring omitted. Then by putting the engine on first one dead center and then the other, two vertical lines may be made on an indicator card to mark the extremes of travel of the

drum. When the engine is run, a horizontal line will overtravel the vertical ones, as was previously demonstrated, if the speed is enough to cause any inertia effect. The overtravel at the crank end should not be greater than 0.01 in. for each foot of cord, the cord being of good quality. If the overtravel exceeds this, the spring should be tightened, until it is reduced to the named amount.

The spring should not be any tighter than necessary, as this would increase the reaction on the drum spindle and therefore the friction. For different speeds, different spring tensions should be used.

(e) **Testing Drum Motion with the Drum Motion Tester.** The apparatus shown by Fig. 36 was devised by the author for

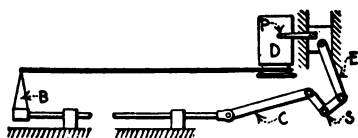


FIG. 36.—Smallwood's Drum Motion Tester.

this purpose. A shaft *S*, the rotary speed of which may be controlled by a variable speed motor, actuates two similarly proportioned crank trains *E* and *C*, set exactly 90 degrees apart. The motions of the two cross-heads are thus exactly proportional at all parts of their strokes. The indicator drum *D* is oscillated by fastening its cord to a bracket *B* carried on an extension of the horizontally moving cross-head. The cross-head with vertical travel carries a pencil point *P* which traces a diagram on a card on the drum. A diagram thus obtained is one of cross-head motion shown vertically and drum motion horizontally. If a rigid connector between the drum and the bracket were used instead of the indicator cord, the motion of the drum would be exactly proportional to that of the pencil, and the diagram

would be an inclined straight line. The effect of an elastic connector, as cord, variously stressed, is to give a curved line. If a straight line is drawn between the highest and lowest points of this curve, then the horizontal departure of any point on the curve from the straight line shows the error in the drum motion at that point.

When testing a drum motion for errors in an indicator diagram previously obtained, great care should be taken to reproduce all the operating conditions, relative to speed, spring tension, length of diagram, length of cord, general adjustment, and

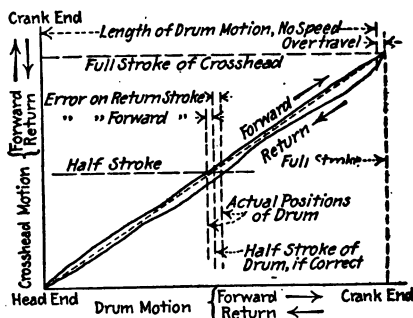


FIG. 37.—Error Diagram from Drum Motion Tester.

friction. Special care should be taken to reproduce the arrangement of guide pulleys. The effect of guide pulleys, especially if near the indicator, is similar to that of drum spindle friction and imposes an additional tightening force on the forward stroke and slackening one on the return. Fig. 37 shows a typical error diagram taken with the drum motion tester and is self explanatory.

In some cases the drum advances its correct position instead of lagging behind it on the return stroke. This is caused by the departure of the acceleration force from the variation assumed in Fig. 35, because of abnormal stretch of the cord and the consequent change in the drum velocities.

(f) **The Correction of Indicator Diagrams from Drum Motion Tester Records** may be accomplished as shown by Fig. 38, which needs no comment. Sampling of the diagrams may be done as indicated under Test 10 (e).

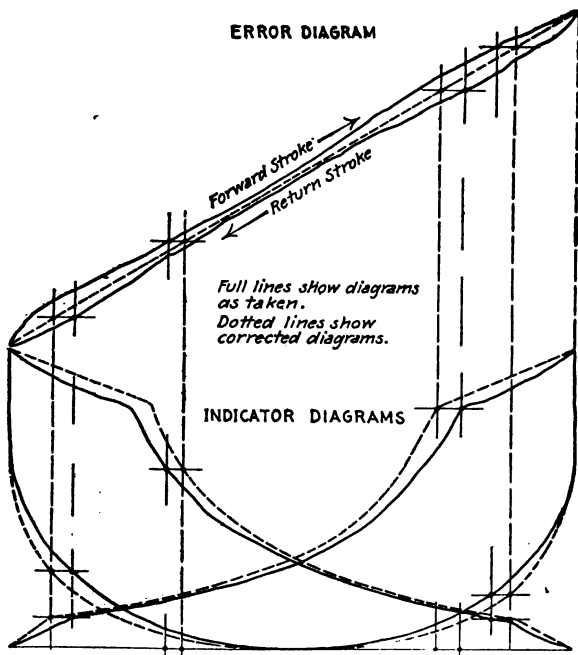


FIG. 38.—Correction of Indicator Diagrams.

**Problem 11.** The force of acceleration of the indicator drum varies directly with its stroke and the square of the engine R.p.m. If a drum motion with a 4-in. stroke is designed correctly to operate at 200 R.p.m., how long should the stroke be to operate correctly at 250 R.p.m.? At 300 R.p.m.?

Ans., 2.56 in.; 1.78 in.

**Problem 11.** What should be the constant of a drum spring (pounds per inch of extension measured on the card) to balance throughout a 4-in. stroke a force of acceleration which has a maximum value of +1 lb. at the

head end, and  $-1$  lb. at the crank end? If the engine R.p.m. is doubled, what should be the constant of the spring? *Ans.*, 0.5 and 2.0 lbs. per in.

## 12. THE TESTING OF LINK TYPE REDUCING MOTIONS

**Principles.** The pantograph in various forms has been much used to reduce engine cross-head motion for the purpose of driving indicators. Fig. 39 shows a number of them, diagrammatically. In these and in the following two figures, the letter  $F$  denotes the fixed center of the linkage;  $R$ , the point at which the indicator cord is attached; and  $C$ , the point of attachment of the linkage to the engine cross-head.

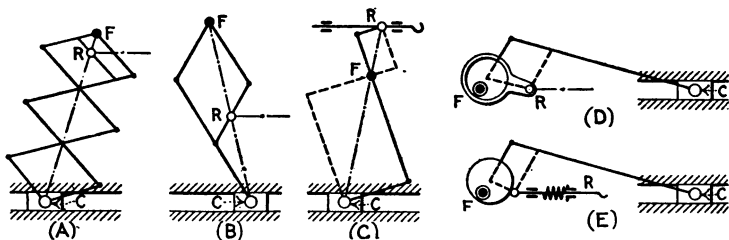


FIG. 39.—Pantograph Reducing Motions.

There are two conditions necessary to the proportionality of motion of the points  $R$  and  $C$ . First, they must lie on a straight line passing through  $F$ , and second, the links that are parallel in one position must remain so in all positions.

Fig. 39 A and B are familiar forms, the reduction of motion being in the proportion of  $FR$  to  $FC$ . The identity of Fig. 39 C may be established by the dotted lines, the linkage thereby represented being replaced by the sliding bar on which the point  $R$  is centered. With this arrangement, the ratio of the long to the short link below  $F$  must be equal to the corresponding ratio above, in order to fulfill the condition of parallelism. This reducing motion is appropriate to high speed engines since by it only



a short length of cord need be used. Fig. 39 *D* is a convenient form of pantagraph made by attaching to the engine crank shaft a small eccentric and rod, or crank and rod, the throw of which is equal to the desired drum stroke. It will be seen that for a correct reduction, the eccentricity must be in line with the engine crank; and the ratio of the lengths of eccentric rod to eccentricity must equal the ratio of the engine connecting rod to crank length. Fig. 39 *E* is a modification of this, the motion being the same as that of a Scottish yoke, that is, a crank train with infinitely long connecting rod. The motion is therefore inaccurate.

In each case, the proportional motion of *R* is *parallel* to that of *C*. Therefore, the indicator cord should be led from the reducing motion parallel to the cross-head guides; if on a slant, the drum motion will not be proportional to the cross-head motion.

Reducing motions of the pendulum type are represented by Fig. 40. By *A* is shown the slotted pendulum. The shorter

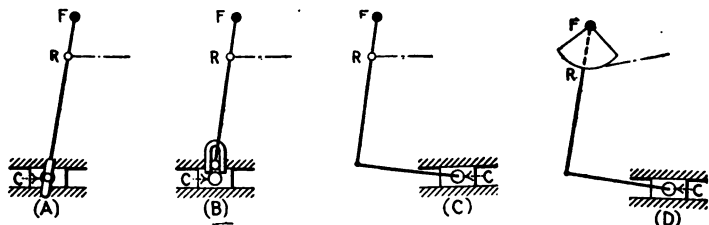


FIG. 40.—Pendulum Reducing Motions.

is the indicator cord, the greater will be its angularity due to the circular motion of *R*. The motion of *R* is not truly proportional to *C* since the ratio of *FR* to *FC* varies. Fig. 40 *B* shows the slotted cross-head, a correct motion except for the angularity of the cord. Fig. 40 *D* is the same as *C* except that a "brumbo" pulley is attached, by which the cord may be led off at an angle with less error than if the pulley were not used. With a horizontal



Care should be taken that motion is not lost through stretch of the indicator cord. For reliable results, wire should be used between the reducing motion and the indicator drum. If the wire is not sufficiently flexible to pass around the drum, a few inches of cord tied to the wire may be used for this purpose.

**Problem 12.** The stroke of the drum given by the reducing motion of Fig. 39 *E* is 4 ins. The ratio of the engine connecting rod to crank length is 5 : 1. Figure from the kinematic formulas the maximum error in the drum motion.

**Problem 12.** Is the best arrangement of Fig. 40 *C* with the lower link always above the horizontal, always below it, or partly above and partly below? Why? What effect has the length of this link upon the accuracy of the motion?

**Problem 12.** Draw two indicator diagrams, superimposed, to show the effect of twisting the eccentric of Fig. 40 *D* ahead of the crank by 10 degrees.

### 13. THE TESTING OF REDUCING WHEELS\*

**Principles.** Link reducing motions have been largely supplanted by reducing wheels on account of the latter's ready adaptability. This type of motion consists, in general, of two drums of different diameters mounted on separate shafts that are connected together with gears. Fig. 42 shows them on the same shaft for simplicity. The indicator cord is led from the engine cross-head to the larger drum to which it is fastened. Another cord connects the smaller drum to the indicator drum. It will be seen that the velocity of the one cord is to the other as the ratio of the drum diameters of the reducing motion. The reducing wheel is supplied with a spring which acts the same as the indicator drum spring. One of the reducing motion drums is made interchangeable with others of various diameters so that different reductions may be made.

\* For a more complete discussion of this subject, and experimental results, see author's article in *Power*, Sept. 24, 1912.

In the operation of a reducing wheel the same forces are at work as in the case of the indicator drum (Test 10, principles), namely, spring tension, friction, and the force due to inertia. Since the indicator drum and wheel masses are connected by a short cord having inconsiderable stretch, they may be regarded as one mass producing a single inertia effect. Likewise, the

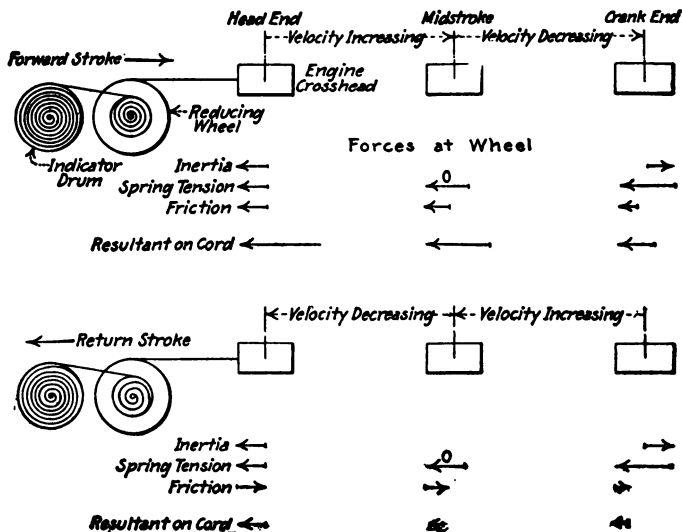


FIG. 42.—Forces on Cord Driving Reducing Wheel.

two springs may be considered as exerting a single force. In Fig. 42, the component forces are shown as they are felt at the cord at three parts of each forward and return stroke. The arrows represent roughly by their length the comparative magnitude of the forces.

At the beginning of the forward stroke, the cord exerts a maximum force in overcoming friction, inertia and spring ten-

sion, but as most of the cord is wound on the wheel, its stretch and the consequent lag of the drum are inappreciable.

As the cross-head advances, the length of cord free to stretch becomes greater, but, as the velocity of the moving parts approaches its maximum in the neighborhood of mid-stroke, the effect of inertia is to lessen the force exerted by the cord and therefore its stretch. This continues throughout the forward stroke; as the length of the cord increases its stress decreases, and the net effect is to maintain a reasonably uniform total stretch and consequently to give a correct drum motion. On the return stroke, however, the spring tension is required to accelerate the wheel sufficiently so that the cord does not slacken, and it must do this not only against inertia, but against the frictional resistance.

As the friction is large, the spring may not be able to do this at a rate equal to that at which the cross-head is being accelerated, particularly at the beginning of the stroke, as here the acceleration is greatest. The stretch of the cord is therefore released at the beginning of the return stroke, if not entirely slackened, as evidenced by whipping. Beyond mid-stroke, the moving parts of the wheel have gained sufficient momentum to draw up the cord again, and this momentum keeps it taut to the end of the return stroke. Under ordinary circumstances, then, a marked distortion of the drum motion may be expected only through the return stroke and greatest at its beginning.

Obviously the length of the cord free to stretch, i.e., that part of it not wrapped around the wheel, has a decided influence upon the correctness of the motion. In practice this length at its minimum and maximum is determined by the engine stroke. Ordinarily, at the head end, the length of cord free to stretch is about equal to the stroke, and at the crank end twice as long.

Experiments upon reducing wheels will disclose the following characteristics.

*First.* The overtravel of the indicator drum is small at both ends of the stroke.

*Second.* The distortion of the return-stroke motion is very marked and follows a characteristic wave. The forward motion is in all cases very closely true.

*Third.* The piston speeds producing whipping and therefore limiting the operation of the wheel, are much greater at the longer strokes.

The first effect is largely due to friction which helps the springs to check the velocity of the wheel at the end of the forward stroke. Much overtravel at the crank end would be evidenced by whipping of the cord. For instance, if the drum motion were 4 ins. and the engine stroke 24 ins., an overtravel of the drum of  $\frac{1}{8}$  in. would mean a slackening of the cord six times that amount, or  $\frac{3}{4}$  in., since any distortion of the drum motion is multiplied at the wheel by the ratio of the motions. This is enough to cause whipping unless the cord is very stretchable. At the head end the overtravel is subdued not only by friction but by the positive resistance of the cord, the stretch of which is a minimum at this end because of its reduced length.

Overtravel of the drum motion does not indicate error with reducing wheels, as it is always small, exists for even low speeds when the motion is actually accurate, and may not exist at all when the friction is such as to cause large error.

The second effect is due not only to friction, but is increased because an indicator cord varies very considerably in length for changes of stress below 1 lb., although above that amount the stretch is comparatively small and uniform with increasing stress. Also the cord will contract markedly when the stress reduces to zero, and sometimes has been observed to continue contracting after the stress was entirely removed. If the frictional resistance to motion of the reducing wheel prevents the spring from properly accelerating it upon the return stroke, the cross-head motion gains upon the wheel motion, thus reduc-

ing the stress in the cord, perhaps to only a few ounces. The cord accommodates itself to this change in length because it contracts markedly at low stresses, and not until the distortion of the drum motion is quite pronounced is the speed of contraction exceeded by the cross-head speed, as evidenced by whipping.

The lag of the drum motion just after passing the crank-end dead center is only momentary at moderate speeds because the force required to accelerate the wheel is a maximum at dead center; when it has once started upon the return stroke less force is needed to continue the motion and this the spring is able to deliver. At higher speeds the action of the spring is further delayed, so that the late acquired momentum of the parts causes them to overshoot the mark and fetch up against the positive resistance of the cord, resulting in the wave effect.

The third effect follows from the limitations caused by inertia. It may be explained by the fact that at a given piston speed the oscillations per minute of the wheel are greater the smaller the engine stroke, and therefore the effect of inertia is greater with the small strokes. Obviously, it is more difficult to oscillate a mass moving at a fixed average speed, the more frequent are the oscillations. This also follows from the mathematical value for the force required to accelerate a reducing wheel at the ends of the stroke (assuming harmonic motion) which may be expressed as follows:

$$A = W \times L \times N^2 \times \text{a constant}$$

in which

$A$  = Force of acceleration referred to the cord;

$W$  = Weight of moving parts referred to the cord;

$L$  = Length of engine stroke;

$N$  = Revolutions per minute.

In this expression  $LN$  is proportional to the piston speed. The force  $A$  cannot be greater than a certain value which limits the

operation of the wheel. The product  $LN$  will have a maximum value, corresponding to the limiting value of  $A$ , by increasing  $L$  rather than  $N$ , since  $A$  varies as the first power of  $L$ , and as the square of  $N$ .

This analysis does not consider the inertia of the indicator drum. The throw of the drum is practically constant at all strokes, so the force required to accelerate it at the ends of the stroke is

$$a = w \times l \times N^2 \times \text{a constant}$$

or

$$a = w \times N^2 \times \text{a constant}$$

in which  $w$  is the weight of the drum and  $l$  its stroke. Thus,  $a$  varies with the square of the R.p.m. only.

The total force of acceleration of drum and wheel is the sum of these forces  $a$  and  $A$ , but as the drum spring is strong enough to meet the drum inertia at the highest speed of oscillation possible with the usual reducing wheel, the force  $a$  generally need not be considered.

If the wheel velocities are low so that inertia is not material, the frictional resistance to motion, if excessive, may cause a material lag of the drum behind its correct position throughout both strokes, since it effects a stretch of the cord upon the forward and a slackening upon the return stroke.

If the variation of stress in the cord is the same for different engine strokes, the distortion of the drum motion is the same. This is because any error in the motion of the wheel is reduced at the drum in the ratio of the drum travel to the wheel travel. For example, if the stresses are the same, the stretch of the cord on a 4-ft. engine stroke is double that of a 2-ft. stroke, since the cord is twice as long, but as the resulting drum distortion is  $\frac{1}{2}$  of the stretch in the one case (drum stroke being equal to 4 ins.) and  $\frac{1}{4}$  in the other, the effect is the same.



**(a) Determination of Spring Tension and Friction.** The method is the same as described under Test 10 (a) and (b). The friction in a well designed wheel is not likely to be less than 20 per cent of the spring tension, and may be as much as 50 per cent in poor designs or when the reducing wheel is badly assembled. As the greatest source of error is in friction, this test is an important one and furnishes a good indication of the merit of a reducing wheel.

When the wheel spring tension cannot be made great enough to accelerate the parts correctly, the deficiency may be made good by tightening the indicator drum spring, to a limited extent only. The drum spring acts upon the wheel at a mechanical disadvantage on account of the reduction of motion, and therefore has only a small effect to accelerate it. On the other hand, tightening the drum spring increases the reaction on the drum spindle, and therefore the friction to be overcome throughout the stroke, and this may cause even more distortion. As a general rule, it is best that the indicator drum spring have only just sufficient tension to overcome its own inertia. The wheel spring tension should be as high as possible if its ability to accelerate the wheel parts is doubted.

**(b) Testing Reducing Wheels with the Drum Motion Tester.** To adapt this device, Fig. 35, to reducing wheels, a bracket, or some equivalent attachment is necessary with a long stroke to duplicate an engine cross-head motion; and, to make the apparatus cover all operating conditions, the stroke should admit variation. This is conveniently accomplished by coupling an extension rod to the motion tester, carrying at its outer end a rack meshing with a pinion on a vertical shaft. This shaft *S*, Fig. 43, carries a wooden disc replaceable by others of different diameters. When the rack is reciprocated, the disc is given an oscillating motion. An extension of the indicator cord from the reducing wheel being fastened to a point on the circumference of the disc, the cord will be reciprocated through a stroke

the length of which depends upon the diameter of the disc. By using different discs, a reducing wheel may thus be operated through any desired stroke.

The reducing wheel to be tested is connected to an indicator and the whole set in place on the tester. The wheel may then be operated at any required conditions of stroke, speed, and cord length; and error diagrams similar to Fig. 37 taken. The cord length may be fixed by tying the desired length to a piece of stranded wire the other end of which is attached to the disc

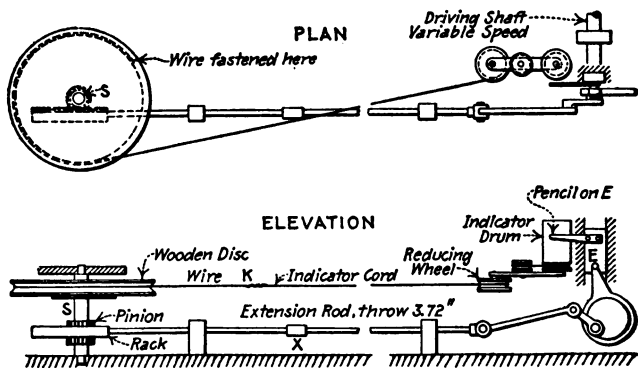


FIG. 43.—Smallwood's Drum Motion Tester Applied to Reducing Wheels.

of the tester. The wire being practically inextensible, the effect is to reciprocate the cord through the desired lengths as by an engine cross-head at K, Fig. 43.

Fig. 44 shows a number of error diagrams taken in this way.

(c) **Correction of Indicator Diagrams.** This may be done as described under Test 11 (f). The reducing wheel is reasonably accurate on short engine strokes and moderate piston speeds, and on long engine strokes at all usual piston speeds *if properly designed to avoid friction*. The drum motion resulting from its

use has its maximum error, when the friction is small, at a part of the stroke where usually it would affect the indicator diagram little if any, namely, at the beginning of the return stroke where either the steam line on the crank end or the exhaust line

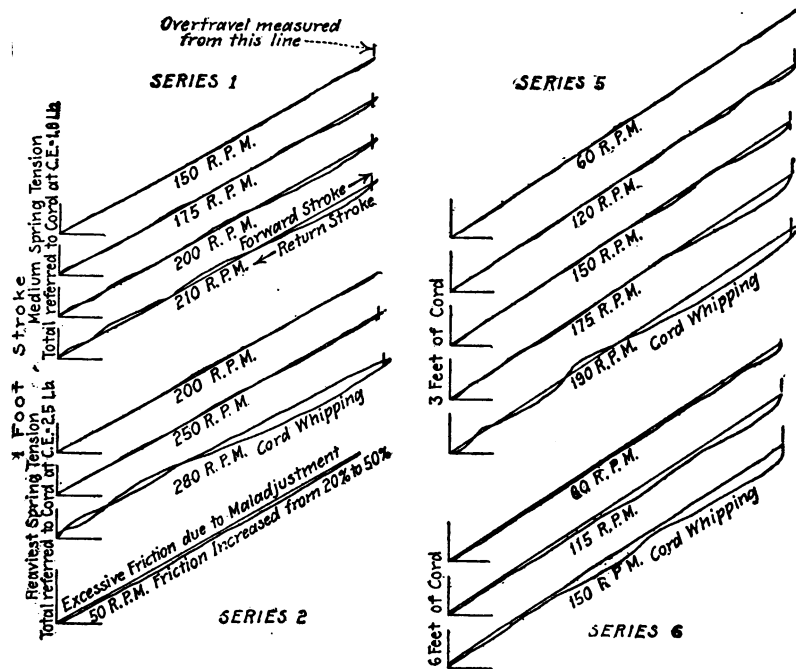


FIG. 44.—Error Diagrams from Reducing Wheels.

on the head end diagram is described. As these lines are generally horizontal, or nearly so, errors upon them would not appear, except at their ends.

## IRREGULAR AREAS AND MEAN HEIGHTS

Numerous engineering instruments, of which the engine indicator is one, have been devised to give an autographic diagram of the measured quantity expressed as an ordinate, the abscissæ being generally time or in linear units simply. From such a diagram it is often desired to get the mean value of the measured quantity, that is, the mean ordinate of the diagram to scale. For this purpose, *planimeters* are used. Some planimeters measure the mean ordinate directly; others measure the area of the diagram, from which the mean ordinate may be found by dividing by the length. The best known of these two types are the Amsler Polar planimeter and the Coffin averaging instrument. The former is used to measure any irregular area; the latter is applied chiefly to the determination of the mean height of indicator diagrams.

### 14. THE POLAR PLANIMETER

**Principles.** Fig. 45 shows in diagram the Amsler planimeter. When the tracing point traverses any closed curve, 1-2-1 the

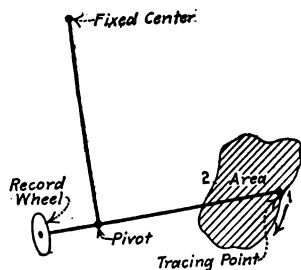


FIG. 45.—Polar Planimeter.

record wheel follows in a certain path and, through contact with the surface upon which it bears, is given a motion partly rolling and partly sliding. The principle upon which the instrument depends is that the rolling motion of the wheel is directly proportional to the area circumscribed by the tracing point. If, then, the wheel is properly graduated, the area may be read directly.

When the area to be measured is comparatively large, the



terms of the constants of the instrument. The motion is that which takes place when the tracing point has outlined the differential of area. 3. By comparing the two expressions (for the area and for the motion of the wheel) the desired relation is obtained.

Fig. 47 shows the differential of area, marked 4-5-6-7. Using the notation of the figure,

$$\begin{aligned}\text{Area 1-4-5} &= \frac{1}{2}r \cdot rdK = \frac{1}{2}r^2 dK \\ \text{Area 1-7-6} &= \frac{1}{2}r_1^2 dK\end{aligned}$$

$$\text{Subtracting, Area 4-5-6-7} = \frac{1}{2}dK(r^2 - r_1^2). \quad . \quad . \quad . \quad (2)$$

This is the expression for the differential of area.

Consider now the corresponding motion of the record wheel. It is to be observed first that this is not affected by the radial motion of the tracing point. The angle between the arms when the point is at 4 (Fig. 47) is the same as that at 5. The same applies to points 6 and 7. Hence, the motion of the wheel when the tracing point passes on the radial from 5 to 6 is the same as that when it passes from 7 to 4. But as these two motions are opposite, they neutralize each other. The same reasoning applies to any irregular area as in Fig. 45. The radial

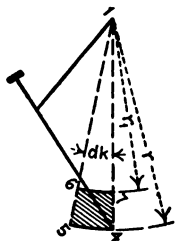


FIG. 47.

component of the motion from 1 to 2 is the same in amount as that caused in passing from 2 to 1, but opposite in direction. So we may altogether disregard the motion of the wheel produced by the radial motion of the point.

Considering again the differential of area of Fig. 47, it is seen that the motion of the tracing point from 4 to 5 is greater than that from 6 to 7, and that the angle made by the arms is different when the point traverses the two arcs. Hence the circular component of the motion of the tracing point when traversing a closed curve produces a record on the wheel. We have, then, to consider the effect of this component.

See Fig. 48. The record wheel moves from 2 to 2' when the tracing point passes from 4 to 5. The rectangular component of the motion, 2-2', causing rotation of the wheel, is represented by the line  $m$ , perpendicular to the wheel axis.  $m$  is therefore

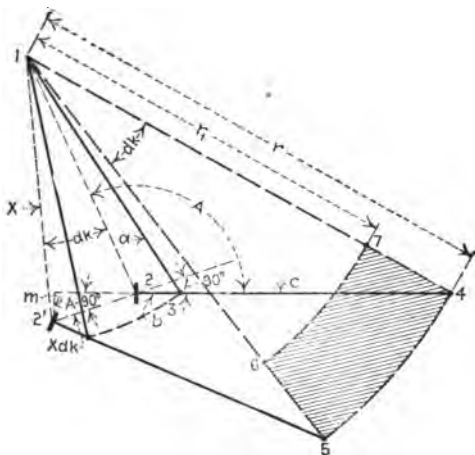


FIG. 48.

the distance moved through by any point on the record wheel circumference relative to its axis.

$$m = X dK \sin (A - 90^\circ) \\ = -X \cos A dK \quad . \quad . \quad . \quad . \quad . \quad (3)$$

From 1-2-3,

$$a^2 = b^2 + X^2 - 2bX \cos A.$$

From 1-2-4,

$$r^2 = (b+c)^2 + X^2 - 2(b+c)X \cos A.$$

Subtracting,

$$a^2 - r^2 = -2bc - c^2 + 2cX \cos A.$$

From which,

$$X \cos A = \frac{1}{2c}(a^2 + c^2 + 2bc - r^2).$$

From (1),

$$= \frac{1}{2c}(r_0^2 - r^2) \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Combining (3) and (4)

$$m = \frac{dK}{2c}(r^2 - r_0^2), \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

which is the value for the rotation of a point on the circumference of the record wheel when the tracing point moves from 4 to 5. Similarly, when it moves from 6 to 7, the motion is

$$m_1 = \frac{dK}{2c}(r_1^2 - r_0^2).$$

The difference between these two motions,  $m$  and  $m_1$ , is the resultant motion  $M$  of a point on the circumference of the record wheel when the whole area has been circumscribed, or

$$M = m - m_1 = \frac{dK(r^2 - r_1^2)}{2c}. \quad . \quad . \quad . \quad . \quad (6)$$

Comparing (2) and (6)

$$\begin{aligned} \text{Area 4-5-6-7} &= \frac{dK}{2}(r^2 - r_1^2) \cdot \frac{c}{c} \\ &= Mc, \quad . \quad . \quad . \quad . \quad . \quad (7) \end{aligned}$$

which is the required relation.

The deduction of the relation for the second case, namely, when the fixed center is within the area to be evaluated, is similar. Under this condition, however, the circular component of the tracing point is always in one direction. Hence there is no subtraction as indicated by equation (6) and the differential expression is

$$M = \frac{dK}{2c}(r^2 - r_0^2).$$

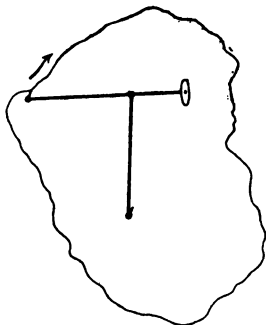


FIG. 49.



Integrating this for the entire circumferential motion of a point on the circumference of the record wheel,

$$M = \int_0^{2\pi} \frac{dK}{2c} r^2 - \int_0^{2\pi} \frac{r_0^2 dK}{2c} = \int_0^{2\pi} \frac{d(\text{area})}{c} - \int_0^{2\pi} \frac{r_0^2 dK}{2c}$$

$$= \frac{\text{area}}{c} - \frac{2\pi r_0^2}{2c}$$

From which,  $\text{Area} = Mc + \pi r_0^2$ , . . . . . (8)

which is the required relation.

From these two deductions it is seen that,

*First.* When the fixed center of the planimeter is without the area to be integrated the motion of a point on the circumference of the record wheel, relative to its axis, multiplied by the length of the arm  $c$ , equals the desired area.

*Second.* When the fixed center of the planimeter is within the area to be integrated, this product added to the area of the zero circle equals the desired area.

In some types of polar planimeter, the arm  $c$  is made adjustable in length, so that areas drawn to various scales may be read directly.

(a) **Determination of the Zero Circle.** If the lengths of the arms,  $a$ ,  $b$ , and  $c$ , are carefully measured, the radius of the zero circle may be computed by means of the relation given by equation (1), and from this the area.

If the axis of the wheel is in a perpendicular plane through the arm  $c$ , the graphic method suggested by Fig. 46 may be used. Two lines are drawn at right angles to each other, the fixed center placed on one, the tracing point on the other, and the point of contact of the wheel and the paper at the intersection of the lines. The radius of the zero circle is then the distance between the fixed center and the tracing point. If the axis of the wheel does not lie in a vertical plane through the arm

$c$  (as is sometimes the case) the construction must be altered to allow for this difference.

**Another method consists in comparing the indications of the planimeter with the calculated area of a circle larger than the zero circle, the arms of the instrument being clamped for convenience when describing the circle with the tracing point.** If the indication of the planimeter be subtracted from the calculated area, the result will be the area of the zero circle. This follows from equation (8).

**(b) Comparison of Instrument Indications with known Areas.**

For this purpose may be used a check rule, an instrument by which the tracing point of the planimeter may be swung through a circle of known radius. Initial and final readings of the record wheel are taken; the difference between these readings should equal the area traversed. When the fixed center of the planimeter is inside the area, the difference between the readings should be added to the area of the zero circle.

If the instrument indications do not correspond to the actual areas, the length of the arm,  $c$ , should be adjusted.

Notice that the wheel reads positively if the direction of the tracing point is clockwise, with the single exception of the case when the area to be integrated is less than the zero circle and the fixed center is inside the area.

**(c) Comparison of Instrument Indications with Areas to Scale.** Planimeters with adjustable arms generally are arranged to be applicable to various scales. For instance, for a certain adjustment, each graduation of the wheel may indicate one square foot on an area drawn to a linear scale of  $\frac{1}{4}$  in. equals 1 ft. For comparison the check rule may be used, its area in square feet on a  $\frac{1}{4}$ -in. scale being figured by multiplying its area in square inches by the square of the linear scale, that is, 16.

**(d) Arm Adjustment to a Required Scale.** Suppose it is required to read areas to a scale not provided for by the markings of the adjustable arm. The necessary length of the arm  $c$

may be calculated from equation (7). In this equation, we may assume the area to be one square inch. Then the motion  $M$  equals the length of one graduation on the wheel multiplied by the number of graduations previously selected to represent the scale units in one square inch.

Let  $w$  = diameter of record wheel, inches.

$G$  = number of graduations on the record wheel.

$X$  = number of graduations representing one scale unit of area.

$Y$  = number of scale units of area in one square inch.

Then, length of one graduation =  $\frac{\pi w}{G}$  in inches.

Graduations representing the scale units in one square inch =  $XY$

$$M = \frac{\pi w}{G} XY.$$

From (7)  $\frac{\pi w}{G} XYc = 1 \text{ sq. in.},$

in which everything is known except  $c$ , which may be found.

A convenient application of this principle is in the direct determination of horse-power from indicator diagrams, for which purpose the arm  $c$  should be adjusted to the length given by the following formula.

$$c = \frac{10500lG}{SKwX}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

in which  $G$ ,  $w$ , and  $X$  are as previously defined, and

$l$  = length of the indicator diagram, inches,

$S$  = scale of the indicator spring,

$K$  = product of  $LAN$  in the  $PLAN$  formula (see Test 41).

**Problem 14<sub>1</sub>.** Deduce the equation for the zero circle of a planimeter having the record wheel between the pivot and the tracing point instead of as shown by Fig. 46.

**Problem 14<sub>2</sub>.** If  $a$  is 5 ins.;  $b$ , 2 ins.; and  $c$ , 3 ins.; what is the area of the zero circle applicable to an arm adjustment for a  $\frac{1}{2}$  in. = 1 ft. scale?

*Ans.*, 2340 sq.ft.

**Problem 14<sub>3</sub>.** The fixed center of a planimeter is inside an area to be integrated which is smaller than the zero circle; consequently the wheel rotates backward. The first reading is 29.23 sq. ins.; the second, 48.73 sq. ins. What is the area if the zero circle area is 212 sq. ins.?

*Ans.*, 292.5 sq.in.

**Problem 14<sub>4</sub>.** Integrate the area enclosed by a loop like a figure 8, first by finding separately the areas of the loop, and second by tracing the loop in the direction of its curve. Account for differences.

**Problem 14<sub>5</sub>.** The arm  $c$  of a planimeter is 2.0 ins. long and the diameter of its record wheel is 0.79 in. How many turns will the wheel make when the tracing point circumscribes 117 sq. ft. to a  $\frac{1}{2}$  in. = 1 ft. scale?

*Ans.*, 13.3.

**Problem 14<sub>6</sub>.** If the wheel diameter is 0.79 in., what should be the length of the arm  $c$  so that one revolution corresponds to 1000 square miles on a linear scale of  $\frac{1}{2}$  in. = 1 ft., there being 100 graduations? *Ans.*, 6.25 in.

**Problem 14<sub>7</sub>.** Deduce equation (9).

**Problem 14<sub>8</sub>.** Find the mean height of a given indicator diagram with a planimeter. Calculate the horse-power. Find the horse-power by adjusting the arm as described under (d). Compare results.

## 15. THE COFFIN PLANIMETER

**Principles.** Fig. 50 represents this planimeter. It consists of a single arm one end of which carries the tracing point, the other end bearing on a pin which slides in a guide represented by the line  $gg$ . The record wheel is mounted on the arm as shown. An examination of Figs. 50 and 45 will show that the Coffin planimeter is the same in principle as the polar; the mechanical difference being that the arm  $a$  of the former is infinitely long, so that the pivot swings in a right line instead of the arc of a circle. The relation  $Mc = \text{area}$  therefore applies (see Test 14, principles).

The instrument may be used for finding areas, but its chief use is in determining the mean heights of indicator diagrams.



diagram, and  $c$  the length of the planimeter arm (see Test 14, principles). Also,

$$\text{area} = h \times l$$

in which  $h$  is the mean height and  $l$  the length of the diagram. Hence

$$Mc = hl, \text{ from which } h = \frac{Mc}{l}.$$

From the similarity of triangles 1-3-4 and 6-7-5,

$$\frac{S}{M} = \frac{c}{l}, \text{ from which } S = \frac{Mc}{l}.$$

Hence  $S = h$ , that is the distance 1-2 equals the mean height of the indicator diagram.

(a) **Comparison of Records with Known Mean Heights.** Instead of an indicator diagram, a carefully laid out rectangle may be used. The tracing point should be started in the lower left-hand corner, and the figure traced back to this starting point. The parallel motion necessary to bring the wheel back to its initial reading will take the point back to the upper right-hand corner of the rectangle, if the instrument indicates correctly. This test should be repeated several times, care being taken that the point pursues the path of the rectangle closely.

If the instrument does not indicate correctly, it may be because the length of the arm has been altered, through bending of the tracing point or otherwise, because of faulty graduations of the wheel, or because of the wheel sticking instead of revolving freely.

**Problem 15.** If the length of the arm  $c$  is 5 ins., what should be the diameter of the wheel so that one revolution corresponds to 10 sq. ins. of area?

*Ans.*, 0.636 in.

**Problem 15.** If the test under (a) gives results uniformly 8 per cent. too small, is the arm too long or too short, and how much should it be changed if its length is 6.12 ins.

*Ans.*, 0.45 in.

**Problem 15.** Using the indicator diagram of Problem 14, find the mean effective pressures by the Coffin planimeter, and compare results with those from the Amsler.

## FLUID VELOCITIES—METERS

Meters for the measurement of fluid quantities are of two broad types: those that measure volume or weights directly and those that measure velocities.

**Volume meters** are so arranged that all of the measured fluid passes through them, alternately filling and emptying compartments and thereby displacing a moving part which registers through a gear and counter combination, the quantity passed. This type of meter gives the total quantity at any time. To get rates it is necessary separately to count the time.

**Velocity meters** measure quantity by means of the relation that the flow in volume units per unit of time is equal to linear velocity multiplied by the cross-sectional area of the fluid. Such meters are dependent upon the uniformity of velocity throughout the cross-section, or upon the correctness of an estimated average velocity when it is not uniform. They may be calibrated to give volumes or weights per unit of time. To get total quantities, it is necessary to multiply by the time.

### 16. CALIBRATION OF A VOLUME WATER METER

**Principles.** In some types of water meter, the moving part is a piston or disc which is displaced by the water entering the compartment of which this moving part is a wall. There may be leakage past the moving part or the valve motion which controls it. This leakage will increase with the pressure drop through the meter. The greater the pressure urging the water through the meter, the faster is the flow. Hence, the accuracy of a water meter will vary with the rate of flow. Other types of water meter, which may be arranged to avoid such leakage, still will have variable accuracy owing to the effect of inertia, etc., at different rates. For a complete calibration, then, a meter should be tested at enough different rates to cover its range.

(a) **Calibration against a Calibrated Tank or Scales.** The meter is arranged so that the water passing through it can be weighed in a tank placed on a platform scales or measured in volume from the known dimensions of the tank. If the volume is measured and the instrument reads in pounds, or vice versa, it is necessary to know the density of the water with reference to its temperature. Enough readings of both the true quantity as shown by the scales or tank and of the meter should be taken to get fair values of the rates of flow. In this connection, rules 4 and 5 given on page 8, should be carefully followed. From the observations, should be figured a series of "true rates" in pounds, gallons, or cubic feet per unit of time, and of "rates by the meter" in the same units. If desired, these may be plotted for a calibration curve.

(b) **Curve of Correction Factors.** This is more convenient to use with a volume meter than a calibration curve, because the instrument is not as a rule used to get rates, but total quantities. The correction factor at any rate of flow is that number by which the total quantity as shown by the meter for any length of time is multiplied to get the true quantity. Consequently, the correction factor is the ratio of the true rate to that shown by the meter. A series of values of the factor may be calculated from the calibration curve and plotted against the rate by the meter.

In using the curve, to select the appropriate correction factor, a rough value of the rate by the meter is figured, and from the curve the corresponding factor is obtained. The true quantity for the total time is then readily determined.

**Problem 16.** Discuss the following observations taken from the test of a meter, and from them figure the true rate, the rate by the meter, and the correction factor.

Time.	True Weight.	Weight by Meter.
2:10	50.5 lbs.	10.0 lbs.
2:15	68.8	20.8
2:20	80.1	31.5
2:25	91.3	42.1



The first true weight is that of an empty tank on a platform scales.

*Ans.*, 2.25 and 2.13 lbs. per min.; 1.06.

**Problem 16.** If a water meter of the volume or displacement type is accurate at all rates when the water is 60° F., draw its calibration and correction factor curves to be used when the water is at 120° F.

## 17. CALIBRATION OF A VOLUME GAS METER

**Principles.** The gasometer is the most accurate instrument of this type. It is represented by Fig. 52, and consists of two

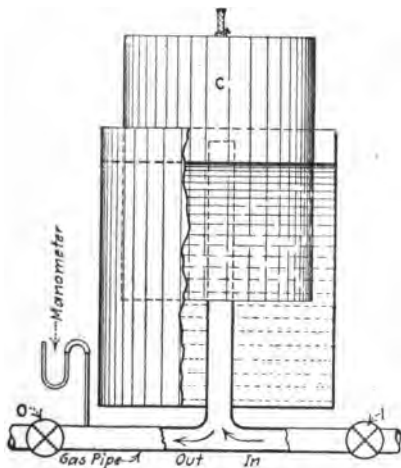


FIG. 52.—Gasometer.

tanks as shown, the upper one being movable vertically and properly counterbalanced. This arrangement makes a chamber of variable size and water sealed. When the upper tank is raised, gas is drawn through the inlet pipe, the valve *O* being closed and *I* open. When lowered, the gas is discharged, the valve control being reversed. The volume of gas thus displaced is measured

by the vertical motion of the upper tank, its cross-section being known.

It will be noted that the gasometer cannot be used for measuring continuous flow unless two are operated, one to fill and one to discharge, alternately.

Another form of gasometer is shown by Fig. 53. Gas is drawn into the cylindrical chamber *C* by allowing water to flow out through the outlet pipe. This gas is then displaced through the

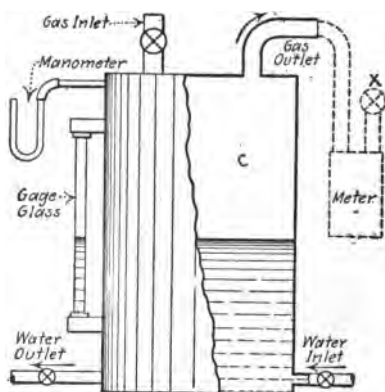


FIG. 53.—Gasometer.

gas outlet by causing water to enter through the water inlet, the valves being properly adjusted. The gas displaced is measured by the rise of water level in the gage glass.

Commercial forms of gas meter are generally of the “dry meter” type and are arranged to measure continuous flow. In this type there are two bellows chambers which are alternately filled and emptied. One side of each bellows being stationary, the other one is thus given a motion which actuates through a linkage the valve control and the recording mechanism.

As gas meters in general record volumes under the existing conditions of pressure and temperature, these conditions should be noted if it is desired to translate the readings into weights or into volumes under standard conditions. (See p. 154.)

(a) **Calibration against a Gasometer.** The meter to be tested is arranged as shown by the dotted lines of Fig. 53. The gas used for testing may be air; the gas inlet may then draw through an open pipe. The gasometer being full of air, water is caused to enter through the water inlet, its rate of flow being the desired gas rate. This is obtained by manipulating the water inlet and noting the time of rising in the gage glass. The valve in the outlet pipe at the meter is then adjusted so as to give a constant air pressure as shown by the manometer which, if equal to that usual in city gas mains, should be about four inches of water.

For other details, see Test 16 (a).

(b) **Curve of Correction Factors** is obtained the same as for a water meter, Test 16 (b).

**Problem 17<sub>1</sub>.** In calibrating a gas meter the capacity of which is 500 cu. ft. per hour, what rates should be applied, and how long should each trial be so that the error due to reading the gage glass is no more than 1 per cent? The diameter of the gasometer is 3 ft.

**Problem 17<sub>2</sub>.** What should be the capacity of a gasometer to calibrate a meter of 2000 cu. ft. per hour capacity?

## 18. CALIBRATION OF A WEIR

**Principles.** A weir is a water meter of the velocity type. It is formed by a notch made in a dam like obstruction in a stream of water through which notch all of the water is caused to flow. The level of the water behind the dam stands above the bottom edge or sill of the notch, and according to a definite difference in these levels, a definite velocity is attained by the water, and hence the quantity passed in a unit of time. If it is arranged

that the difference of level be measured by a precise instrument, then the flow may be calculated; or, if the weir has been calibrated, the flow may be obtained from the calibration curve.

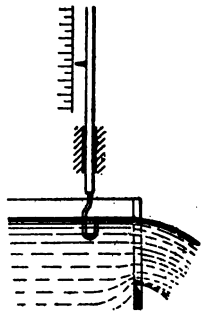


FIG. 54.  
Weir and Hook Gage.

Fig. 54 shows a weir, the height of the water level above the sill being represented by  $H$ . The pressure to which the particles of water at the sill are subjected is equal to  $H$  ft. of water. When this pressure is reduced to zero by emergence of the water into the atmosphere, the work done is  $WH$  ft.-lbs. per second,  $W$  being the weight of water in pounds per second. This work is transformed into kinetic energy, the expression for which is  $\frac{WV^2}{2g}$ ,  $V$  being the

velocity in feet per second, and  $g$  the acceleration of gravity. Consequently,  $WH = \frac{WV^2}{2g}$  and,

$$V = \sqrt{2gH}.$$

The velocity of the particles of water at levels above the sill is less than this since they are at less pressure, and it may be shown that the average velocity is  $\frac{2}{3}V$  or  $\frac{2}{3} \times \sqrt{2gH}$ . If the breadth of the weir is  $B$  ft., then the cross-sectional area to which this average velocity applies under ideal conditions is  $BH$  sq. ft. Then, since

$$\text{Quantity} = \text{area} \times \text{velocity},$$

it follows

$$\begin{aligned} Q' &= BH \times \frac{2}{3} \sqrt{2gH} \\ &= \frac{2}{3} B \sqrt{2gH^3}, \end{aligned}$$

$Q'$  being in cubic feet per second.

This is for ideal conditions. Actually, the velocity is somewhat less than the ideal because of eddies and friction of which the theory takes no account. So we may write

$$\text{Actual velocity} = C_1 \times \text{ideal velocity.}$$

$C_1$  is less than unity and is called the "coefficient of velocity."

Also the actual water section is less than  $BH$  since there is generally a contraction due to a fall at the top and a narrowing at the edges of the weir on account of the tendency of the water at the sides to continue in the plane of the weir. Hence,

$$\text{Actual area} = C_2 \times BH$$

in which  $C_2$  is the "coefficient of contraction."

It follows that the actual quantity

$$Q = \frac{2}{3} C_1 C_2 B \sqrt{2gH^3},$$

$$Q = \frac{2}{3} C B \sqrt{2gH^3},$$

$C$  being called the "coefficient of discharge."

For rectangular weirs with sharp edges,  $C$  equals about 0.62 depending upon the size of the weir and the head.

When the stream behind the weir is small in cross-section, its velocity is relatively high. The so-called velocity of approach will then appreciably increase the velocity of the water through the weir since it acts in addition to gravity, and should be allowed for. The average velocity of approach may be calculated by multiplying the velocity through the weir by the ratio of water sections at the weir and in the flume behind the weir. The additional head urging the water through the weir is then

$$\frac{(\text{Velocity of approach})^2}{2g}$$

which may be added to  $H$  for use in the formula for  $Q$ . This is an approximation since the velocity in the flume is very variable throughout the cross-section. The actual velocity of approach is greater since the water in the middle of the stream, having greatest influence on the passage through the weir, is of higher velocity than at the sides and bottom. It is more correct, then, to multiply the velocity head as just given by a number greater than unity which, according to Hamilton Smith, lies between 1.0 and 1.5.

Many modified formulas have been proposed for weirs to allow for the variations of  $C_2$  and  $C$ . The most notable, perhaps, is Francis'

$$Q = 3.33 (B - 0.2H) \sqrt{H^3},$$

in which 3.33 equals  $C \frac{2}{3} \sqrt{2g}$ . In this, it is regarded that each end contraction increases with the head and equals  $0.1H$ . There is thus less variation in the applied value of  $C$  which remains approximately equal to 0.62. This formula may be used for uncalibrated weirs having  $B$  greater than 4 ft. when the head,  $H$ , exceeds 5 ins.; with less than 1 per cent of error.

Other shapes of weir notches, the coefficients for which are not so well established as for rectangular weirs, are the V-notch and the trapezoidal. For the former the equation is

$$Q = C \frac{8}{15} \sqrt{2g} H^{\frac{5}{2}}$$

when the sides of the notch make a right angle.

There is less variation in the coefficient of contraction for this type of weir than the rectangular. Also it is useful for variable flow and small quantities since the head diminishes at a rapidly decreasing rate with the quantity.

Merriman recommends an average value of  $C$ , 0.592, which gives

$$Q = 2.53 \sqrt{H^{\frac{5}{2}}}.$$

V-notch weirs are now made with elaborate indicating and recording apparatus to register the flow.

The trapezoidal weir may be arranged so that the extra breadth due to the slope of the sides as the head rises balances the increased contraction of section so that the product of the breadth  $B$  at the sill and the coefficient of contraction remains nearly constant. The slope of the sides should then be 1 to 4. The value of  $C$  may be taken as 0.62, and we have, as for a rectangular weir without end contractions,

$$Q = 3.33 B \sqrt{H^3}.$$

The difference of level,  $H$ , is generally measured by the hook gage, Fig. 54, by which minute differences of level may be detected. In use, the hook is started just below the water level, then raised slowly until it barely breaks the water surface, fastened in that position and then read at the graduated scale.

(a) **The Zero of the Hook Gage** is its reading at a given datum plane. For a weir, this plane is the horizontal one through the lowest edge of the notch. To get the zero of the hook gage for a weir, a straight edge should be placed with one end at the lowest point of the notch and the other end on the point of the hook, and so balanced that very little weight comes on the hook, to prevent springing it. The hook is then raised or lowered until a spirit-level placed on the straight edge shows the horizontal, and a reading taken. The straight edge and level are then turned end for end and the procedure repeated. By averaging the two readings, any untruth of the straight edge or level is eliminated.

(b) **To Calibrate a Weir**, a series of readings of head and corresponding quantity rates in cubic feet per minute or second should be obtained and plotted. To get the quantity rate, the water may be discharged into a calibrated tank, the rate of rising in which will give the volume per minute. When the weir is at the end of a flume of uniform horizontal cross-section, a convenient

method is to start with it empty, and then adjust the incoming water to the desired rate of flow. The rate can be ascertained as the flume is filling by readings of the hook gage taken at equal increments of time. When the water level has reached its highest position, uniform flow through the weir being established, a final reading of the hook gage gives the head corresponding to the rate thus previously ascertained. With this method, it is especially important that the rate be constant toward the end of each run.

(c) **The Coefficient of Discharge** varies with the rate. For any rate, it may be found from the quantity formula, the values of  $H$  and  $Q$  being known.

**Problem 18.** Find the coefficient of discharge from the following experimental data obtained from a rectangular weir. Size of flume 18 ft. long by 3 ft. wide. Rate of rise, 0.12 ft. in 5 sec. Head on the weir, 5 ins. Breadth of weir, 18 ins. *Ans.*, 0.6.

## 19. CALIBRATION OF A NOZZLE

**Principles.** A nozzle makes a very convenient form of velocity water meter when the water to be measured may be discharged from a pipe into the atmosphere. The principle upon which this meter depends is similar to that of a weir, that for a definite pressure behind the opening there will be a definite flow through it. Then, if it is arranged that this pressure be measured, the flow may be ascertained in cubic feet or pounds per unit of time either by calculation or from a calibration curve.

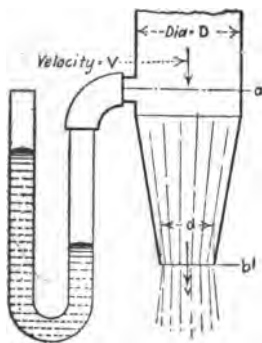


FIG. 55.—Nozzle.

The equation of a nozzle may be obtained from a consideration of the energy appearing in the water. In Fig. 55 the energy at



the section  $a$  is in two forms, pressure and velocity, neglecting the small amount of potential energy due to the height at  $a$  above the opening at  $b$ . All the energy at  $a$  is converted into velocity at  $b$ , since the pressure at  $b$  is nil, except that used to overcome fluid friction between the points  $a$  and  $b$ . Let  $W$  equal the weight of water passing per second.

Then the available energy at  $a$  is  $WH + \frac{WV^2}{2g}$ ,  $H$  being the pressure in feet of water and  $V$  the velocity in feet per second at  $a$ .

The expression  $\frac{WV^2}{2g}$  is the kinetic energy at  $a$ , and  $WH$  the pressure energy. Likewise, the energy delivered at  $b$  is  $\frac{Wv^2}{2g}$ .

If  $v_1$  is the velocity at  $b$  under the ideal condition of no losses, then

$$Cv_1 = v,$$

in which  $C$  is a coefficient less than unity. Since energy is proportional to the square of the velocity,

$$C^2 \times \text{available energy} = \text{delivered energy}.$$

Substituting in this the values of the energies as previously noted,

$$C^2 \left( WH + \frac{WV^2}{2g} \right) = \frac{Wv^2}{2g}.$$

We also have the relation that the velocities are inversely proportional to the cross-sectional areas, or

$$\frac{V_1}{v} = \frac{\frac{\pi d^2}{4}}{\frac{\pi D^2}{4}} = \frac{d^2}{D^2}.$$

Combining these last two equations we have

$$v^2 = \frac{C^2 2gHD^4}{D^4 - C^2 d^4}.$$

Since the quantity,  $Q$  in cubic feet per second, equals the velocity times the area of the stream,

$$Q = 0.7854d^2 \sqrt{\frac{C^2 2gHD^4}{D^4 - C^2 d^4}}.$$

For convenience, this may be expressed

$$Q = 6.3C \frac{D^2}{\sqrt{R^4 - C^2}} \sqrt{H},$$

in which  $R$  is the ratio of  $D$  to  $d$ . Note that the units of  $D$  are feet.

The value of  $C$  for a well designed nozzle is between 0.95 and 0.99. It is thus seen that such a nozzle may be used as a meter without material error if it is uncalibrated.

A convenient method of measuring the pressure is by a mercury manometer as shown in Fig. 55. The difference in level of the mercury in inches should then be multiplied by  $\frac{13.6}{12}$  to convert into feet of water. The lower level of mercury should be referred to the datum plane through the end of the nozzle (when the nozzle is vertical) as this allows for the potential energy between the sections  $a$  and  $b$  which was neglected in the formula; the column of water in the right-hand leg of the manometer balancing that within the nozzle. If the mercury descends below the datum plane, the reading of the manometer must be corrected for the head of water between the lower mercury level and the datum. When the nozzle is horizontal, the datum plane should be through the axis of the nozzle.

(a) **Calibration at Various Rates.** The rate may be varied by turning a stop valve in the pipe to which the nozzle is affixed, and the rate may be measured by discharging the water into a calibrated tank or by weighing it. If the pressure measuring device is a manometer, the tube connecting it with the nozzle should be full of water as any air in it will cause the apparent water head to be different from the actual by amounts varying with the pressure. The water head below the datum plane may be corrected for by translating it from inches of water to inches of mercury and subtracting from the manometer reading; or the manometer may be raised each time it is read so that the lower mercury level is at the datum plane at the instant of reading. The determinations of rate in the desired units should be plotted against pressures in inches of mercury.

Rules 4 and 5, page 8, should be carefully observed.

(b) **The Coefficient of Discharge** varies somewhat with the rate, at any value of which it may be found from the quantity formula, the values of  $H$  and  $Q$  and the diameters of the nozzle being known.

**Problem 19.** What is the discharge in cubic feet per second from a nozzle having a diameter ratio of 2 to 1, the smaller diameter being  $\frac{1}{2}$  inch, if the manometer shows 5 ins. of mercury with its lower level 7 ins. below the nozzle mouth? Assume  $C=0.97$ . *Ans.*, 0.0246 cu.ft. per sec.

**Problem 19.** Supposing the water in the above to be at  $60^{\circ}$  F., what will be the difference in the discharge, expressed in cubic feet and in per cent, if the water is at  $120^{\circ}$  F., assuming  $C$  the same? The conversion figure 13.6 is used in both cases. *Ans.*, 0.4%.

## 20. CALIBRATION OF A VENTURI METER FOR WATER

**Principles.** The venturi meter is similar in principle to the nozzle. It is, in fact, a nozzle discharging into a closed and properly shaped pipe instead of into the atmosphere. See Fig. 56. The water passing through the pipe shown carries a certain

amount of energy in the form of pressure and velocity. When the water reaches the contracted section *b*, its velocity is increased,

and therefore its pressure must be decreased, since the total energy, barring losses, remains constant. The drop in pressure between sections *a* and *b* thus becomes a measure

of velocity and hence quantity.

To deduce the equation for the venturi meter it is only necessary to equate the expressions for energy at the two sections *a* and *b*. Thus, if *V* and *v* are the velocities in feet per second, and *H* and *h* the pressures in feet of water at the sections *a* and *b*, respectively, and *W* the weight of water passing per second, then

$$WH + \frac{WV^2}{2g} = Wh + \frac{Wv^2}{2g},$$

*g* being the acceleration of gravity. We also have the relation

$$\frac{V}{v} = \frac{\text{Area at } b}{\text{Area at } a} = \frac{d^2}{D^2}.$$

Combining these equations and solving for *v*,

$$v = \frac{D^2}{\sqrt{D^4 - d^4}} \sqrt{2g(H - h)},$$

from which the cubic feet per second discharged under ideal conditions is

$$\begin{aligned} Q' &= \text{area} \times \text{velocity} \\ &= 0.7854d^2 \frac{D^2}{\sqrt{D^4 - d^4}} \sqrt{2g(H - h)}. \end{aligned}$$

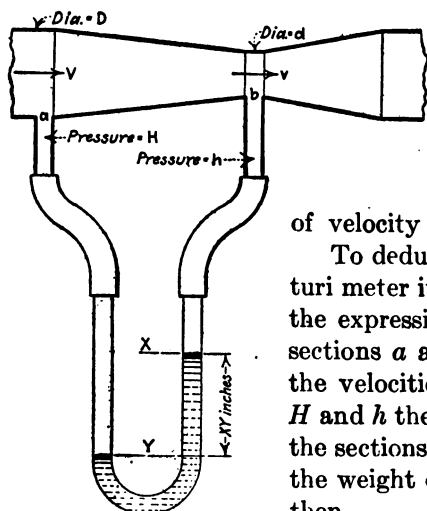


FIG. 56.—Venturi Meter.

Owing to friction losses the discharge thus calculated is too high. To allow for this, a coefficient of discharge,  $C$ , less than unity, is introduced. Combining  $\sqrt{2g}$  and 0.7854, for convenience, the formula becomes

$$Q = 6.3C \frac{D^2}{\sqrt{R^4 - 1}} \sqrt{H - h},$$

in which  $R$  is the ratio of  $D$  to  $d$ , and  $D$  is in feet.

The value of  $C$  for a well designed venturi meter is between 0.95 and 0.99, and the value of  $R$  is usually 2 or 3 to 1. It is thus seen that an uncalibrated venturi meter may be used with not more than 2 per cent of error by assuming  $C = 0.97$ .

For measuring the difference of pressure,  $H - h$ , a mercury manometer is generally used connected as shown by Fig. 56. When the manometer is graduated in inches, the flow may be calculated or obtained from a calibration curve determined experimentally. The manometer is sometimes graduated in quantity rates so that no curve need be used. Some forms of venturi meter employ apparatus giving a continuous record on a time chart which, when integrated, yields total quantities.

The meter is generally shaped with an annular space  $A$ , Fig. 57, from which to lead the opening transmitting pressure. Between this annular space, or pressure chamber, and the interior of the tube are circular openings connecting the two. This arrangement transmits the pressure more truly than would a single opening.

(a) **The Calibration at Various Rates** may be made by controlling the rate by a stop valve in the water line and measuring the rate by discharging the water into a calibrated tank or by

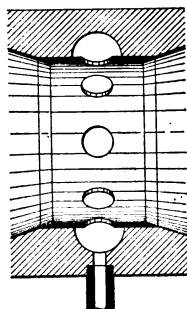


FIG. 57.

weighing it. A curve should be drawn between the rates and the differences of mercury level (manometer readings).

Rules 3, 4 and 5, page 8, should be carefully observed.

(b) **The Coefficient of Discharge** is determined from the formula at any rate when the values of  $Q$ ,  $H-h$ , and the diameters of the instrument are known.

The head in inches of mercury should be transposed to feet of water. Referring to Fig. 56, it is seen that the column of water on the right above  $X$  balances an equal column on the left. The column of water between  $X$  and  $Y$ , however, is balanced by mercury, so that the pressure difference between sections  $a$  and  $b$  is not  $XY$  inches of mercury, but  $XY$  inches of mercury minus  $XY$  inches of water; that is

$$\frac{(13.6-1)}{12}XY = 1.05 XY, \text{ feet of water.}$$

**Problem 20<sub>1</sub>.** Draw the calibration curve for a venturi meter having a diameter ratio of 3 to 1, the smaller diameter being 1 in., after obtaining rate values upon the assumption that  $C=1$ .

**Problem 20<sub>2</sub>.** Assuming that the energy lost in the venturi meter varies as the square of the velocity, draw a curve between rates and corresponding values of  $C$ , the minimum value of  $C$  being 0.97. Proportions of meter the same as in preceding problem.

## 21. CALIBRATION OF A VENTURI METER FOR GAS

**Principles.** The deduction of the flow formula differs from that for water on account of the fact that gas carries intrinsic energy due to its expansive property which must be accounted for in the equation of energy. Otherwise the principle of the deduction is the same. The flow formula has been presented by Mr. E. P. Coleman in a paper to be found in the transactions of the A.S.M.E., Vol. 28, page 483. The general arrangement of the instrument is the same as for water. It has not been generally used for gases probably because the pitot tube (see

Test 22) answers the same purpose more cheaply, is considerably smaller, and has been more developed experimentally.

(a) **Calibration at Various Rates** may be accomplished by using one of the methods described under Tests 23, 25, 26, or 27 for measuring the true quantity of gas; or if the venturi meter is of small size, by using a gasometer. The rates may be obtained in cubic feet per second or minute, and plotted against difference of pressure in inches of water or mercury. If the weight of gas is desired, temperatures should be noted. Note that the temperature conditions must not materially vary from those in use, since a change of temperature is accompanied by a change of density of the gas.

(b) **The Coefficient of Discharge** may be calculated from the previously mentioned formula. As this involves the absolute pressures at the two sections of the meter, and not merely the difference of pressure, the pressures should be read separately and the barometer reading added in similar units.

## 22. CALIBRATION OF A PITOT METER FOR WATER

**Principles.** Fig. 58 represents a pitot meter, which is merely a curved tube placed in a stream of velocity  $V$  feet per second so that the immersed end of the tube faces the stream. The kinetic energy of a given weight of the water,  $W$ , is  $WV^2/2g$ . The kinetic energy causes a rise of the water in the tube above the main water level to a height  $H$ , such that the work done in lifting the water to this height equals the kinetic energy. Hence,

$$WH = \frac{WV^2}{2g}$$

and

$$V = \sqrt{2gH}.$$

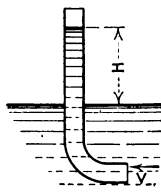


FIG. 58.  
Pitot Tube.

The height  $H$  thus becomes a measure of velocity and therefore quantity.

If the stream is in a closed conduit, the water may be under a pressure greater than atmospheric which would cause it to rise in the pitot tube to a height greater than that due to velocity. This may be allowed for by making a separate measurement of pressure. Thus, in Fig. 59,  $H_2$  ft. of water balances velocity plus pressure, and  $H_1$  ft. in the straight tube is due to pressure only owing to the fact that the velocity is not impressed upon this tube opening. The velocity is then

$$V = \sqrt{2g(H_2 - H_1)}.$$

The pitot tube opening is called the "velocity opening" and the other, the "static opening." They are often connected to a differential gage as shown by the dotted lines of Fig. 59 so that the pressure is balanced, and a single reading gives the velocity head direct.

The plane of the static opening should be parallel to the stream; for, if it is inclined toward it, the velocity is partially impressed; or, if away from it, suction will result. With a correct static opening there may be the same effects if the stream is not parallel to the pipe direction, which may be the case at points near bends or elbows, or when the instrument itself interferes with the regularity of the flow. Generally two static openings on diameters at right angles, connected together, are used.

When the pitot tube and static opening are properly designed and applied, the actual condition of flow is the same as that represented by the formula, so that there need be no coefficient to determine as with the nozzle, weir, and venturi meters.

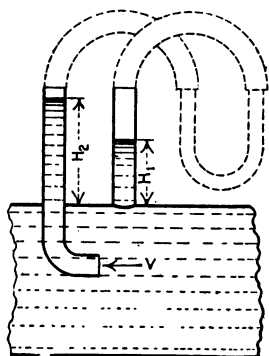


FIG. 59.—Pitot Meter.



In a closed conduit, the velocity of the stream varies through the cross-section, being maximum at the center and minimum at the sides. It is therefore necessary either to find the velocity at enough points to get a fair average or to read the instrument at a point of mean velocity. The mean velocity is approximately  $.7854 \times$  velocity at center, and this rule is sometimes used for rough results.

**(a) Determination of Average Velocity and of Quantity Rates by Traversing.**

In Fig. 60, the concentric circles mark off equal areas so that  $a_1 = a_2 = a_3 = a_4 = a_5$ , the area of the pipe being  $A = 5a_1$ . The average velocities in these areas are  $v_1$ ,  $v_2$ ,  $v_3$ , etc., respectively, the average velocity in the whole pipe section being  $V$ . Then, if  $Q$  represents the cubic feet per second,

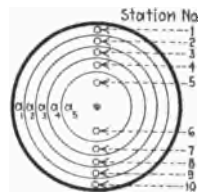


FIG. 60.

$$\begin{aligned} Q &= AV = a_1v_1 + a_2v_2 + a_3v_3 + \text{etc.}, \\ &= 5a_1V = a_1(v_1 + v_2 + v_3 + \text{etc.}), \end{aligned}$$

and

$$V = \frac{v_1 + v_2 + v_3 + \text{etc.}}{5}.$$

That is, the average velocity in the pipe is the average of velocities taken at points representing equal areas.

The method to be pursued, then, is to take readings of the differential gage at such points for a single determination of quantity, the quantity being calculated by taking the product of the average velocity,  $V$ , and the area of the pipe. It is customary to take readings at ten points, or stations, as shown by Fig. 60. The readings on one side of the pipe center should duplicate those on the other, and there are two readings for each area. The stations should be located at the following

distances from the pipe wall,  $D$  being the diameter of the pipe, in order to represent equal areas.

Station No. 1,	.0256D	No. 6,	.658D
No. 2,	.0817D	No. 7,	.774D
No. 3,	.146D	No. 8,	.854D
No. 4,	.226D	No. 9,	.918D
No. 5,	.342D	No. 10,	.974D

The procedure is simplified by taking the average of the square roots of the differential gage readings, and using this in the velocity formula,  $V = \sqrt{2g} \times \sqrt{H}$ . Note that the head of the gage liquid should be converted into equivalent head of water in feet. If mercury is used as the gaging liquid, this may be done as described under Test 21 (b). If some other liquid, as oil, is used, its specific gravity should be taken into account according to the same principle. The quantity rate can be expressed as

$$Q = a \text{ constant} \times \Sigma \sqrt{\text{gage reading}}$$

for the simplification of numerical work.

(b) **Location of the Point of Mean Velocity.** If the pipe is traversed and the average of the square roots of the gage readings obtained as under (a), then the square of this average represents the gage reading at the point of mean velocity. This point may then be located by plotting on a chart gage readings against distances from the wall of the pipe; the distance corresponding to the square of the average square root of the gage readings being then taken from the chart.

The location may be found without plotting a curve by searching with the pitot tube for the point at which is registered the gage reading as calculated above and then measuring the distance of the velocity opening from the pipe wall. If this is done, a center reading should be taken when making the traverse

by which the uniformity of flow may be checked when the searching is done.

**Problem 22<sub>1</sub>.** Deduce the values given in the table under (a) for the distances of the stations from the pipe wall.

**Problem 22<sub>2</sub>.** Give a rough value of the quantity rate in cubic feet per minute in a 10-in. pipe (internal diameter = 10.02 in.) if the differential gage reading at the center is 26 ins. of oil the specific gravity of which is 0.9, an inverted U-tube being used. *Ans.*, 24.2 cu.ft. per min.

**Problem 22<sub>3</sub>.** Figure the constant for use in the quantity formula for an 8-in. pipe (internal diameter = 7.98 in.), the gaging liquid being mercury. *Ans.*, .0714, in cu.ft. per sec. for 10 stations.

**Problem 22<sub>4</sub>.** Figure the cubic feet per second in an 8-in. pipe (internal diameter = 7.98 ins.) if a traverse gives the readings, 4.1, 5.05, 5.7, 6.25, 6.75, 6.6, 6.0, 5.5, 5.0, 4.05 inches of mercury. *Ans.*, 1.66 cu.ft. per sec.

## 23. CONSTANTS OF A PITOT METER FOR GAS

**Principles.** The pitot meter may be used for gas exactly the same as for water (see Test 22) except that greater precautions should be taken on account of the fact that gas is a much more mobile fluid. The instrument should be placed at a distance of at least 12 pipe diameters from the nearest bend on the up-stream side, and four on the down-stream. A pitot tube of unusual proportions or design should not be used without previous calibration as it has been found by experiment that apparently unimportant details cause large errors in the indications. The best proportions of pitot meter for gas under general conditions have not yet been satisfactorily established, but Fig. 61 represents what probably is the best practice at the present time.

In the case of gas, in the formula

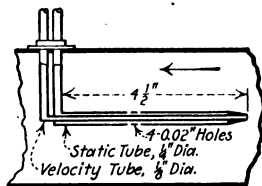


FIG. 61.—Pitot Tube for Air.

$$V = \sqrt{2gH},$$

$H$  is the head due to velocity expressed in feet of whatever gas is flowing, allowing for its density due to its condition of pressure and temperature. The observed head in terms of inches of the gaging liquid must therefore be translated. For this purpose the following relation may be used,

$$H = \frac{h}{12} \times \frac{\text{density of gaging liquid}}{\text{density of gas in pipe}},$$

in which  $h$  is the observed head in the U-tube connected as shown in Fig. 59, expressed in inches. The densities are generally given in pounds per cubic foot. Water is usually used for the gaging liquid, the density of which may be taken as 62.3 lbs., as it varies but little with temperature. Kerosene is also used. Its density may be figured from its specific gravity.

**(a) Determination of Average Velocity and of Quantity Rate by Traversing.** The experimental procedure is exactly the same as for Test 22 (a) when the gas is at room conditions of temperature and pressure, that is, approximately 70° F., and 14.7 lbs. Air, for example, under these conditions weighs .0749 lbs. per cubic feet, so

$$H = \frac{h \times 62.3}{12 \times .0749} = 69.3h,$$

from which

$$V = \sqrt{2g(69.3h)} = 66.7\sqrt{h},$$

in which  $h$  is the observed head in inches of water.

At other pressures and temperatures, the density varies according to the relation (for perfect gases)

$$144pv = p \frac{144}{w} = RT',$$

in which  $p$  = absolute pressure in pounds per square inch,  
 $v$  = specific volume in cubic feet per pound,  
 $w$  = density, in pounds per cubic foot,  
 $R = 53.4$ , for air,  
 $T$  = absolute temperature, degrees F.

From this relation, follows

$$w = \frac{144p}{RT}.$$

Now

$$V = \sqrt{2gH} = \sqrt{2g \frac{h}{12} \times \frac{62.3}{w}}.$$

Substituting the value of  $w$  and simplifying,

$$V = 11.1 \sqrt{\frac{h}{p} T},$$

from which the quantity,  $Q$ , in cubic feet per second, may be figured by multiplying by the area of the conduit.

If the quantity rate, in pounds per second,  $W$ , is wished, the following may be used.

$$W = wQ = \frac{144p}{RT} \times .7854 \frac{d^2}{144} \times 11.1 \sqrt{\frac{h}{p} T},$$

in which  $d$  = diameter of conduit in inches. Simplifying,

$$W = .163d^2 \sqrt{\frac{hp}{T}}.$$

The absolute temperature is figured by adding  $460^\circ$  to the temperature as obtained by a Fahrenheit thermometer. For the absolute pressure of the gas,  $p$ , the barometer should be read and added in the same units to the static pressure of the gas as shown

by a U-tube or other pressure gage. This gage may be connected to the static opening of the pitot meter.

For any other gas than air, or for any other gaging liquid than water, an equation may be deduced similar to the above by the same method.

The procedure, then, for gas at temperatures and pressures materially different from room conditions is either

*First*, to make a traverse from which the average of the square roots of the differential gage readings yields  $\sqrt{h}$ , and to get one set of readings of temperature, static pressure, and barometer.

*Second*, to place the Pitot tube at a point of average velocity, by which  $h$  is obtained by a single reading, other measurements the same.

(b) **The Location of the Point of Mean Velocity** may be found exactly the same as for water, Test 22 (b).

**Problem 23<sub>1</sub>.** What is the velocity, cu.ft. per second, and pounds per second of flow of illuminating gas in a 3-in. pipe (internal diameter = 3.07 ins.) if the reading of the differential gage is 1.5 ins. of oil of specific gravity 0.85? Pressure of the gas is 4 ins. of water; barometer, 29.7 ins. mercury; temperature 55° F.  $R$ , for this gas, is 60.7. *Ans.*, 0.276 lbs. per sec.

**Problem 23<sub>2</sub>.** In the formula  $V = 66.7\sqrt{h}$ , room conditions are assumed, that is,  $p = 14.7$ , temperature = 70°. How much may the actual condition of pressure depart from this to make less than 1 per cent of error when this formula instead of the more accurate one is used? Give answer in terms of inches of water. How much may temperature depart?

*Ans.*, 8.1 in.; 10.8°

**Problem 23<sub>3</sub>.** What is the velocity in feet per second if  $h = 12$  ins. of water, pressure = 4 ins. of mercury, temperature = 120° F., barometer = 30.1 ins. of mercury, the gas being air. Figure answer by both approximate and accurate formulas.

*Ans.*, 231 and 226 ft. per sec.

## 24. CALIBRATION OF AN ORIFICE FOR WATER

**Principles.** If water is caused to pass through an orifice, the pressure behind the orifice being measured, the flow may be calculated. Suppose that the velocity of the water behind the

orifice is negligibly small; then the energy available is  $WH$ ,  $W$  being in pounds per second, and  $H$ , the pressure in feet of water. This pressure energy is expended in imparting velocity, or kinetic energy, to the water upon passing through the orifice, so that, neglecting friction, etc.,  $WH = WV'^2/2g$ ,  $V'$  being the velocity in feet per second, and  $g$  the acceleration of gravity. Hence, under ideal conditions,  $V' = \sqrt{2gH}$ . Owing to friction losses, the actual velocity is somewhat less than this, so that if  $c_1$  is a number less than one,

$$V = c_1 \sqrt{2gH}.$$

If we multiply this by the cross-sectional area of the stream, we shall get the quantity,  $Q$ , in cubic feet per second. Now, this area is less than that of the orifice; the stream being contracted by virtue of the tendency of the water particles to flow in the plane of the orifice; so that if  $c_2$  is a number less than one,

$$\text{area of stream} = c_2 \times .7854 d^2$$

the orifice being circular and  $d$  ft. in diameter. It follows that

$$\begin{aligned} Q &= c_1 c_2 \times .7854 d^2 \sqrt{2gH} \\ &= c \times .7854 d^2 \sqrt{2gH}. \end{aligned}$$

$c_1$ ,  $c_2$ , and  $c$  are called the "coefficients of velocity, contraction, and discharge," respectively. Merriman gives average values for them as .98, .62, and .61 for the case of orifices with sharp edges, that is, orifices in thin plates beveled on the side not touched by the water, so that the water touches only one line of the orifice.

The coefficients vary with the head and with the diameter of the orifice.

In order that the velocity back of the orifice shall be negligible, the cross-section of the stream in the conduit should be at least

one hundred times that of the orifice. If the water conduit is round, this means its diameter should be at least ten times that of the orifice.

The pressure back of the orifice may be measured by a mercury manometer or by a Bourdon gage, or the actual height of water above the orifice may be measured direct with a gage glass, float, or hook gage (see p. 88).

As with a rectangular weir, the coefficients vary considerably with both the head and the size of the opening. The average value,  $c = .61$ , should therefore be taken only for rough results.

The coefficient of contraction varies with the shape of the orifice, one having edges rounded toward the inside giving markedly less contraction.

(a) **Determination of Quantity Rates at Various Heads** may be made by discharging the water into a weighing tank, or by measuring its volume. The quantity discharged in cubic feet per minute should be plotted against the head in the observed units.

(b) **Determination of Coefficients.** The value of  $c$  may be calculated from the formula when the values of  $Q$ ,  $H$ , and  $d$  are known. The coefficient of contraction may be found by measuring the diameter of the stream at the contracted section with a caliper, this section being distant from the plane of the orifice by about a half diameter. Knowing this coefficient, and the coefficient of discharge, the coefficient of velocity is readily obtained.

**Problem 24<sub>1</sub>.** What is the coefficient of discharge, if an orifice discharges 92 lbs. of water per minute under a pressure of 3 pounds per square inch. Diameter of the orifice is 0.6 ins.

*Ans., 0.594.*

**Problem 24<sub>2</sub>.** If the diameter of the contracted section (see Problem 24<sub>1</sub>) is 0.36 ins., what are the coefficients of contraction and velocity.

*Ans., 0.6; 0.99.*



## 25. CALIBRATION OF AN ORIFICE FOR GAS

**Principles.** The deduction of the flow formula differs from that for water on account of the fact that gas carries intrinsic energy due to its expansive property, which must be accounted for in the equation of energy. The theoretically correct flow formula is deduced in various works on thermodynamics. It includes necessarily a coefficient  $c$  to allow for contraction and losses, as is the case with water passing through an orifice. For gas, however, the value of  $c$  is very uncertain, especially for high pressures, there being insufficient experimental data on this quantity.

When the pressure drop between the two sides of the orifice is small, the volume of the gas changes but little, and consequently only little intrinsic energy is delivered to influence the flow. Under these circumstances, the intrinsic energy may be ignored, and an equation deduced similar to that for water (Test 24, principles), namely  $V = c\sqrt{2gH}$ , the head,  $H$ , then being the height of a column of gas to produce the pressure drop, its density being taken into account.

Since the orifice generally may be designed of such size to produce a small pressure drop, it seems hardly worth while to use the accurate and much more complex formula, especially in view of the uncertainty of the values of the coefficient of discharge under those conditions to which the hydraulic formula does not apply.

The gas may be caused to flow into the atmosphere, in which case, the pressure drop equals the pressure of the gas above atmosphere, and may be read by a manometer. If the gas cannot be discharged in this way, the orifice may be in a plate fitting square across the gas conduit, and the drop of pressure measured with a differential gage as with the venturi meter, Fig. 56, except that water should be used as a gaging fluid.

In any case, the cross-section of the entrance to the orifice should be of such size as to reduce the entrance velocity to a negligible value. If the orifice needs to be large to secure the desired pressure drop, and the conduit is relatively small, the purpose can be accomplished by making an enlarged section in the conduit.

In the case of the pitot meter, Test 23 (a), velocity is measured by measuring a static pressure caused by it. In the case of an orifice, the procedure is reversed in that a static pressure, causing velocity, is measured and from it the velocity is determined. It follows that the derivation of the velocity formula is the same, and we have for an orifice after introducing the coefficient of discharge,

$$V = c66.7\sqrt{h}, \text{ for room conditions of air,}$$

$$V = c11.1\sqrt{\frac{h}{p}}T, \text{ for general conditions,}$$

and for the rate in pounds per second,  $d$ , being the diameter of the orifice in inches

$$W = c.163d^2\sqrt{\frac{hp}{T}};$$

the other notation being as given for the pitot meter.

The value of  $c$  has been fairly well established for orifices in thin plates, up to about 5 ins. in diameter, and to a pressure of about 6 ins. of water. An average value of 0.6 may be taken within these limits with less than 2 per cent of error.

**(a) Determination of Quantity Rates at Various Heads.** The rate may be varied by varying the output or the speed of the machine handling the gas. If an air compressor or blower, the speed may be varied. If the orifice is used for such a purpose as measuring the exhaust from a gas engine, its external load may be varied thus changing the amount of fuel and air used.

The true rate may be measured by any of the methods of Tests 23, 26, 27, or by a gasometer if the volume of gas is not too large.

Readings of the pressure difference between the two sides of the orifice should be plotted in the gage units against true rates.

(b) **Determination of the Coefficient of Discharge.** This is readily calculated from the flow formula, the calibration data of (a) being known. It is instructive to plot values of  $c$  against the pressure drop through each size of orifice, and against orifice diameters for each value of the pressure drop.

**Problem 25<sub>1</sub>.** Write the equation of energy of air on the two sides of an orifice, neglecting kinetic energy at the entrance. Allowing a drop of pressure equal to 6 ins. of water, compare the velocity due to pressure energy with that due to intrinsic energy assuming adiabatic flow.

**Problem 25<sub>2</sub>.** Design an orifice and containing conduit to measure the air supplied to a 12 H.P. gas engine taking about 20 cu. ft. of fuel gas per horse-power hour and about 12 cu. ft. of air to one of fuel. Drop of pressure through the orifice should not exceed 6 ins. of water.

## 26. CALIBRATION OF AN ANEMOMETER

**Principles.** An anemometer is a meter of the velocity type, generally consisting of a wheel with vanes against which the current of gas impacts causing a rotary motion proportional to the velocity of the current. This motion is transmitted to a gear and counter combination which registers linear feet continuously. By counting the time, the velocity in feet per second or minute is calculable.

When used to measure quantity, the anemometer is generally placed at the exit cross-section of the conduit discharging the gas. If the cross-section is rectangular it may be divided into a number of small squares defined, for convenience, by light strings or wire fastened from wall to wall of the conduit. The anemometer is placed in the middle of each of these squares and the velocity read. The average velocity through all of them

may then be used with which to multiply the total area to get cubic feet per minute or second. If the conduit is round, the anemometer should be placed at points located as for a pitot tube, Test 22 (a).

Anemometers generally are not adapted for velocities higher than 100 ft. per second, and are not very reliable. The vanes are apt to become deformed, causing false indications, and changes of frictional resistance of the bearings will have the same result.

When the velocities exceed the capacity of the anemometer, the discharge conduit may be enlarged in cross-section at the exit.

**(a) Calibration against the Velocity of the Instrument in Still Air.** The anemometer is mounted on one end of a horizontal arm 3 or 4 ft. long and pivoted at the other end so that the instrument may be caused to travel through the circumference of a circle. For this purpose the pivot may be supplied with a grooved wheel whereby the arm may be driven, through a belt, by a motor or by hand. Knowing the revolutions per minute of the arm and its radius to the center of the anemometer, the linear velocity of the anemometer may be calculated. This is the true velocity of the air relative to the anemometer, and corresponds to the instrument reading. A number of such determinations are made at different velocities, and plotted as a calibration curve.

Care should be taken that the velocity is uniform throughout each trial, and that the error of starting and stopping is made sufficiently small. A small lever is generally arranged on anemometers by which the recording mechanism may be thrown in or out of gear. This may in some cases be operated while the instrument is moving.

**(b) Curve of Correction Factors.** Corrections may be figured as quantities in linear feet per minute to be added to or subtracted from the instrument indication for one minute. These

should be plotted against linear velocities as shown by the instrument.

**Problem 26<sub>1</sub>.** A blower discharges 2000 cu. ft. of air per minute through an 8-in. pipe. Design an exit conduit to reduce the velocity a proper amount so that the air may be measured with an anemometer, and show where the anemometer should be placed.

**Problem 26<sub>2</sub>.** If the encircling frame of an anemometer is the same diameter as the opening through which gas is discharged, what should be the area with which to calculate quantity? Examine an anemometer to answer this question.

## 27. TESTING A CALORIMETRIC APPARATUS FOR MEASURING GAS

**Principles.** If gas flowing through a conduit is arranged to be heated or cooled by steam, water, or electric current, then, barring radiation from the conduit, the heat gained by the one medium equals that lost by the other. Thus, supposing the gas to be cooled by water pipes, if

$W$  = weight of gas passing in a given time,

$W_w$  = weight of water passing in same time,

$T_1, T_2$  = initial and final temperatures of the gas,

$t_1, t_2$  = initial and final temperatures of the water,

$C_p$  = specific heat of the gas at constant pressure,

then

$$WC_p(T_1 - T_2) = W_w(t_1 - t_2),$$

from which,

$$W = \frac{W_w(t_1 - t_2)}{C_p(T_1 - T_2)}.$$

It is thus seen that with such an apparatus the air passing in a given time may be measured by weighing the water and taking the temperatures of the gas and water before and after cooling.

If steam is used in the coils, being condensed by the air,

it is necessary to take the temperature of the water discharged and the pressure and quality of the entering steam from which its heat content may be obtained with the steam tables.

If electric current is used, the heat equivalent may be figured from voltmeter and ammeter readings.

The calorimetric method is sometimes useful for measuring large quantities of air.

(a) **Examination of Instruments.** The instruments should be sufficiently precise that the error of reading should be less than 2 per cent of the corresponding factor in the formula. The weight of water may be measured readily with proper precision. The temperatures, however, appear as differences which may be only a few degrees. Hence, thermometers graduated to tenths may be necessary. It is sometimes useful to figure beforehand rough values of weights and temperature differences in order to ascertain the required precision of the instruments.

It is best, when using the apparatus, to take temperatures at various points in the cross-section of the gas conduit, to search for variations.

(b) **Determination of Radiation Correction** may be made approximately by stopping the flow of gas and maintaining the temperature within the conduit at an average value between the limits in actual operation. The heat necessary to maintain this temperature may then be figured for a unit of time. This may be used as a correction if very precise results are desired.

**Problem 27.** What should be the least counts of a voltmeter and ammeter to measure 600 cu. ft. of air per minute under room conditions? The voltage of the line is about 110. What should be the least count of the thermometers? Temp. rise of air to be about 20° F. *Ans.*, about 0.5 amp.

## 28. CALIBRATION OF A STEAM METER

**Principles.** Steam meters are built on the pitot, venturi, and orifice principles. The quantity of steam measured is therefore proportional to the square root of a pressure or difference in pressure. The pressure, varying with the flow, actuates an indicating or recording device. When arranged to give a time chart showing rates and total quantities, the recording mechanism is generally rather delicate and complicated. It should be noted that with such meters, the steam flow is proportional not to the motion of the indicator or height of the chart, but to the square root of these quantities.

Another type of meter uses the orifice principle at a constant difference of pressure and varies in orifice size to allow various amounts of steam to pass. This is accomplished by a float set in the orifice and so shaped that its motion varies the effective area of the orifice. The motion is transmitted to an indicating or recording device. In this case, the height of the diagram obtained may be directly proportional to the steam flow.

Changes in density of the steam, due to difference in pressure or superheat, should be allowed for since the meters generally indicate the quantity in terms of pounds. This may be done by applying different tables, furnished by the makers, to interpret the indications, or by a hand adjustment of the recording mechanism. Variation of density due to wetness may be avoided by passing the stream through a separator just before it reaches the meter.

When steam passes through an orifice, dropping in absolute pressure from  $P$  to  $p$ , the flow increases with the drop until  $p=0.58P$ , which is known as the "critical pressure." For lower pressures than this, there is no increase in the flow.

A convenient method of orifice measurement, when the steam

is discharged at a pressure lower than  $0.58 P$ , depends upon the application of Napier's formula, namely,

$$W = \frac{PA}{70}$$

in which  $W$  is the weight in pounds per second of dry saturated steam at an absolute pressure  $P$  lbs. per square inch, behind the orifice, and  $A$  is the area of the orifice in square inches.

Steam meters generally are not to be depended upon when the flow is very variable or intermittent. This condition may often be remedied by the location of the meter.

The polar planimeter may be used to integrate charts in which the height is proportional to the square of the flow provided there is but little fluctuation. The result is approximate since the average height obtained will be the square root of the average of the squares of the flow.

(a) **Calibration** may be made by comparison with an accurate steam meter, or by direct weighing of the steam. The latter method is the more dependable. All of the steam passed through the meter in a given time should be condensed in a surface condenser and then weighed. A calibration curve may be made for each pressure or condition of superheat, if desired; the quantity of steam being varied by a valve on the outlet pipe from the meter. Generally it is sufficient to make the calibration at one condition of pressure, the steam being saturated. Throttling calorimeter readings should be made to insure the latter condition.

**Problem 28<sub>1</sub>.** Using Napier's formula, what is the flow of steam in pounds per minute through a  $\frac{1}{4}$ -in. diameter orifice, if the gage pressure is 30 lbs.? What is the lowest pressure to which this method applies if the steam discharges into the atmosphere? *Ans.*, 0.0314 lb. per sec.; 25 lb. gage.

**Problem 28<sub>2</sub>.** Using a chart record from an actual steam meter, figure the average rate by taking off velocities at a number of equal time intervals. Figure the average rate from the mean height obtained by a planimeter. Compare results.



## THERMOMETRY

Temperature may be defined as a measure of heat. It is purely relative and is generally dated from the freezing point of water.

The expansion of materials, when heated, is approximately proportional to the heat added. It follows that the increase of length of a material is a measure of heat intensity. Upon this principle, the mercury thermometer was devised, mercury being chosen on account of its expansive properties. The Fahrenheit scale of degrees was chosen in a roundabout way by its inventor who exposed his thermometer first to freezing water and then to water at the boiling point. He marked the stem of the instrument at the level of the mercury in each case, and divided the distance between the marks into 180 equal parts. He continued the scale below the freezing point by 32 of these divisions. The freezing point thus became 32 degrees, and the boiling point 212 degrees. Therefore the Fahrenheit degree may be defined roughly as the increase of temperature corresponding to 1-180 of the total linear expansion of the material chosen for a thermometer between the freezing and boiling points of water.

The material may be liquid, solid, or gaseous. Now, no known material expands uniformly in proportion to the heat added. The departure from uniformity, although slight, is appreciable in accurate work, especially at high temperatures. It follows that the divisions of a mercury thermometer, although absolutely equal, do not measure equal amounts of heat, and a degree of temperature is a quantity varying with its location on the scale. Also different materials expand according to different laws. Consequently, two thermometers of different materials, standing the same at the freezing and boiling points of water, will differ at all other points. It must not be thought

that this is because either one is inaccurate; it is simply that they are different standards. The linear expansion of a particular material, such as mercury, yields an entirely definite, though arbitrary, temperature scale, because the expansive properties of such a material, when pure, are constant.

In a mercurial thermometer, however, the degree is not measured by the expansion of mercury only; the glass containing it also expands. The scale, therefore depends upon the relative expansion of mercury and glass. The latter varies considerably both in manufacture and expansive properties, so that for a standard scale it must be specified with great care.

Owing to the difficulty of obtaining glass of uniform quality, other materials than mercury have been preferred for accurate work. For the expansive material, air has been commonly used on account of its great expansiveness compared with that of the containing material. Nitrogen and hydrogen are now in more extensive use.

Gas thermometers are of two types: constant pressure and constant volume. The former uses the expansive properties of the fluid as in the mercury thermometer, while the latter measures the temperature indirectly by the pressure which varies directly with the temperature according to Boyle's law. The resulting scales are different, but the differences are so slight that for most engineering purposes it is not necessary to discriminate between them. The gas scales are very nearly coincident with each other and with the theoretical "work scale" devised by Lord Kelvin. This latter is the ideal scale in which the degrees stand for equal amounts of heat. The simple mercury scale is lower than the others between the freezing and boiling points of water and higher above the boiling point. The mercury-in-glass scale is higher than the others between the freezing and boiling points.

The nitrogen scale is the one commonly employed in scientific work, but the standard (between the freezing and boiling points)

is that of the constant volume hydrogen thermometer. This was defined in 1887 by the International Bureau of Weights and Measures. The standard instrument is operated at an external pressure of one standard atmosphere, or 760 mm. of mercury, and the hydrogen is maintained at a pressure, when at the freezing point, of one meter of mercury.

## 29. CALIBRATION OF HIGH READING THERMOMETERS AND PYROMETERS

**Principles and Types.** Mercury thermometers are made for temperatures up to  $1000^{\circ}$  F. Their use, however, is limited because of their fragility. Instruments for measuring high temperatures are, in general, termed pyrometers.

A much used form of thermometer is practically the same as a recording pressure gage, being of the **constant volume** type. It is arranged to be mounted at some distance from the point where the temperature is to be measured and is connected thereto by a flexible tube ending in a bulb filled with alcohol or ether which bulb is subjected to the temperature. The pressure thus produced within the bulb is transmitted and recorded in terms of temperature upon a chart.

Another form of expansion thermometer takes advantage of the **relative expansion** of two metals. As they are heated, the motion of one relative to the other is magnified by a gear combination and transmitted so as to rotate a needle over a graduated dial, thus indicating the temperature.

The **thermo-electric pyrometer** is one of the most accurate and convenient instruments for measuring temperature. It depends upon the fact that two wires of different metals having appropriate thermo-electrical properties will generate an electromotive force when connected at their ends, which electromotive force is proportional to the temperature difference between the

ends. If it is arranged that the current be measured with a sensitive galvanometer, placed at any convenient distance from the thermo-electric "couple," we have an indirect measure of temperature. The galvanometer may be graduated in degrees or a calibration curve may be used to translate the electrical units.

For temperatures up to 1200 or 1400° F., the couple is composed of a wire of nickel and another of nickel and chromium. For temperatures up to 3000 degrees, the metals are platinum and platinum with 10 per cent rhodium. The latter combination may be used for the low temperatures, but the former is preferred on account of cheapness. Many other combinations have been used, but those cited above are the most usual.

**Optical pyrometers** are advantageous when the hot body to be measured is inaccessible to a couple or bulb. One of them, the Morse, depends upon a comparison of the brightness of the hot body with that of an electric filament which is caused to glow more or less brightly by varying the current passing through it. Then, by measuring the current, an indirect measure of the temperature is obtained as with the thermo-electric couple. When used to measure furnace temperatures in boiler testing, a thin plate of iron is hung at the required point and its brightness is compared with that of the filament. This type of pyrometer is not very precise because it is difficult to judge exact similarity of brightness.

Another optical pyrometer depends upon the thermo-electric couple principle, but the couple is acted upon by radiant heat concentrated upon it by means of lenses in the form of a telescope.

There are other forms of pyrometers using other principles, but the ones listed above are the principal ones.

(a) **Calibration.** There are three ways generally used to calibrate high reading thermometers and pyrometers: first, by comparison with a standard or secondary standard instrument;

second, by comparison with the temperature of saturated or wet steam determined from its pressure; third, by comparison with temperatures as shown by the melting points of metals.

A secondary standard is established after calibration against a constant volume gas thermometer. For this purpose, mercury thermometers may be used for temperatures up to  $1000^{\circ}$  F. Above, and often below that temperature, a favorite instrument is the thermo-electric couple as it is one of the most accurate and easy to use.

The temperature is varied in a gas or electric furnace in which the bulbs of the standard instrument and the one to be tested are placed. Care should be taken to bring the whole furnace up to a uniform temperature before recording observations, by allowing sufficient time for heating. It is best to take a series of decreasing readings to supplement the increasing ones, allowing the furnace to cool for this purpose. If the up and down readings are different at a given temperature, as they will be with certain types of instrument, they should be averaged. (See Test 2, principles.)

In the case of thermo-couples, readings should be made of the temperature of the "cold junction," that is, the ends of the couple to which the galvanometer leads are joined. This is because the temperature indicated by the galvanometer is the difference between those of the cold and hot junctions. The cold junctions are often placed in ice to keep them at a constant known temperature. If a couple has been calibrated in this way against a standard couple, allowance should be made for the temperature of the cold junction when in ordinary use. This may be done by adding to the indications the difference between freezing temperature and the temperature of the cold junction.

The method of calibration by steam is limited by the available pressure. At 200 lbs. gage, the temperature is  $388^{\circ}$  F., and at

500 lbs., about 470 degrees. It is thus apparent that not very high temperatures can be reached with saturated steam.

The procedure is to vary the temperature in a steam chamber similar to that used for testing indicator springs, Fig. 31, page 50, by controlling the pressure. The thermometer is inserted in a well set in the chamber and readings taken of it and of an accurate and sufficiently precise pressure gage at each variation of the pressure. The actual temperatures are ascertained by reference to the steam tables. A barometer reading is necessary to get absolute pressures.

When the steam is throttled to reduce its pressure it may become superheated, in which case its temperature is not determinable. To avoid this, water may be sprayed into the steam before it enters the steam chamber, or the chamber may be water-jacketed. It is essential that there should be definite knowledge of the steam's wetness, for which purpose a throttling calorimeter (see Test 30) may be referred to.

A calibration of an instrument indicating beyond the available temperatures may be had by extrapolation. That is, after the experimental data have been plotted, the curve is continued beyond the known points. If the curve is a straight line, the result may be good, but caution should be used not to extend the extrapolation too far.

The third method of calibration, by the melting points of metals, depends upon the accuracy with which these points are established and the purity of the metals used. The method is perhaps more refined and laborious than is generally necessary in engineering experimentation.

**(b) Stem Corrections.** If a mercury thermometer is to indicate correctly, all of the mercury, both in the stem and bulb, should be subjected to the measured temperature. In practice this is rarely the case, as part of the stem containing mercury is at or near room temperature. For accurate work it is then necessary to apply a "stem correction."

Let  $S$  = stem correction in degrees,

$H$  = height of column above actual position of mercury,  
if uniformly heated, inches,

$l$  = length of one degree on the stem, inches,

$N$  = number of degrees on the stem exposed to room  
temperature,

$T$  = temperature, Fahrenheit, of the bulb,

$t$  = average temperature, Fahrenheit, of the exposed  
mercury and stem,

$C$  = cubical coefficient of expansion of the glass and  
mercury, combined by subtracting the value of  
one from that of the other.

Then

$$H = C(l \times N)(T - t).$$

Also

$$S = \frac{H}{l}.$$

Hence

$$S = CN(T - t).$$

The value of  $C$  for the Fahrenheit scale may be taken as 0.000088 and for the Centigrade scale, 0.00016.

It is generally sufficiently accurate to take the actual reading as the value of  $T$ . If desired, the corrected temperature thus obtained may then be used as  $T$ , and a second and more nearly exact stem correction calculated.

The value of  $t$  is somewhere between room temperature and that of the bulb. It is often estimated by hanging a second thermometer against the one to be corrected, with its bulb at a middle point on the exposed mercury column. It is very doubtful that this yields a close value for  $t$ .

Whether or not a stem correction is necessary depends upon the purpose for which the temperature is measured. In engineering work, thermometers are generally used to measure temperature differences, so that the percentage of error in the result

may be based upon a much smaller quantity than the actual number of degrees read. In many cases, when the temperatures are comparatively low or when the stem is well immersed, the correction is negligible.

**Problem 29.** With a thermometer immersed to the  $60^{\circ}$  (F.) mark, how high a temperature may be reached so that neglecting the stem correction will cause less than 1 per cent of error? With a thermometer to be used at about  $600^{\circ}$  F., to what mark should the stem be immersed to limit the error to the same amount? Take  $t$  as the average between the room temperature,  $60^{\circ}$ , and that indicated. Base percentages upon the maximum temperatures.

First *Ans.*,  $229^{\circ}$ .

**Problem 29.** What should be the least count of a pressure gage to calibrate by steam a thermometer whose least count is 2 degrees?

*Ans.*, 2 lbs. at  $300^{\circ}$ .

**Problem 29.** Account for the differences between up and down readings in the calibration of a thermo-electric couple.

## HEAT OF STEAM--CALORIMETRY--SAMPLING

The English measure of heat is the British thermal unit, or B.t.u. This is loosely defined as the amount of heat necessary to raise the temperature of 1 lb. of water  $1^{\circ}$  F. As the specific heat of water varies with the temperature, a precise definition should name the temperature range. Several ranges have been used, as 32 to 33 degrees, 39 to 40 degrees, 61 to 62 degrees, so that there are a number of values for the B.t.u. The present tendency favors the "mean B.t.u." which is the average amount of heat per degree required to raise a standard pound of water from the freezing to the boiling point. One advantage of this unit is that it is independent of the temperature standard (whether gas or mercury) since it is fixed by the two temperatures which are the same on all thermometric scales. This is the unit used in the Marks and Davis' steam tables, values from which will be used in the examples in this work.

The heat of steam is counted from a temperature of  $32^{\circ}$  F.



and is the amount of heat necessary to convert 1 lb. of water at that temperature into steam under the given physical conditions. This includes the heat added to the liquid to bring it to the boiling point at the given pressure, and the heat then necessary to vaporize it. If the steam is superheated, still another amount of heat is added to it to raise its temperature from that at the boiling point, the pressure remaining the same. All of these heat quantities depend upon the pressure of the steam. It is thus seen that the physical conditions of steam determining its heat content are, first, its pressure, and second, its wetness or superheat. For example, the heat of the liquid, referred to as  $h$ , of steam at 14.7 lbs. absolute, is 180 B.t.u., being the heat necessary to raise the temperature of 1 lb. of water from 32 to 212 degrees; 970.4 B.t.u. are then necessary to vaporize it. The heat of vaporization is referred to as  $L$ . Thus the total heat,  $H$ , necessary to make 1 lb. of dry steam at 14.7 lbs. is.  $h+L=1150.4$  B.t.u. If less than the whole pound of water has been evaporated so that it contains  $1-x$  lb. of water as a mixture,  $x$  being the part of a pound actually evaporated, then

$$H=h+xL.$$

This is the generic expression for the heat added to make 1 lb. of wet steam in which  $x$  may have any value up to unity.

For superheated steam the generic expression is

$$H=h+L+C_p(T-t_s),$$

in which  $C_p$  is the specific heat of superheated steam,  $t_s$  is the temperature of the dry or "saturated" steam at the given pressure, and  $T$  is the temperature of superheat. The value of  $C_p$  varies with  $T$  and with the pressure.

These various physical conditions are represented by Fig. 62 which shows 1 lb. of  $H_2O$  in a cylinder with a movable end under a pressure  $P$ .

It should be remembered that the datum temperature of 32 degrees is taken purely as a matter of convenience, and that the actual amount of heat contained by steam, or for that matter, any material, is a relative quantity. The significance of heat quantities so counted enters when we take the difference between two of them representing two conditions of  $H_2O$ . Thus the heat per pound given up by a boiler in making steam is the difference between the heat content of the  $H_2O$  leaving the boiler, counting from 32 degrees, and that of the feed-water entering, counted similarly.

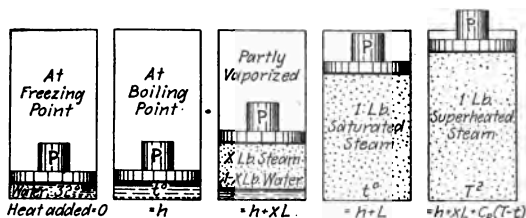


FIG. 62.—Heat Added to One Pound of  $H_2O$ .

To measure the heat of a given quantity of steam, it is necessary to know the amount in pounds and the heat content,  $H$ , or heat added per pound; the one multiplied by the other gives the heat of the total quantity. The weight of steam may be obtained by a steam meter, Test 28, or by any of the methods given under engine or boiler testing. The steam tables are used to get the value of  $H$ , since they list the heats of liquid and vaporization at different pressures. If the steam is dry, it is sufficient to take its pressure or temperature for reference in the tables. If it is superheated, its pressure should be read, and its temperature,  $T$ . The steam tables furnish values of  $t_s$ , and  $C_p$ , from which  $H$  may be calculated, knowing also  $h$  and  $L$ . The steam tables also contain heat contents of superheated steam, so that the calculation is not always necessary.

If the steam is wet, the value of  $x$ , as well as the pressure or temperature, must be known to fix its heat content. This is the condition generally met in practice, which makes determinations of wetness or "quality" important, especially for engine and boiler tests.

**The Heat of Steam when Mixed with Perfect Gases.** If the pressure of the steam were known, its total heat could be found by reference to the steam tables as previously described. When steam is mixed with other gases, however, its pressure is less than that of the mixture. This follows from Dalton's law which may be stated thus,

*In a gas mixture occupying a given volume, the pressure of each constituent gas is the same as though it occupied the volume alone, and the pressure of the mixture equals the sum of the pressures of the constituent gases.*

For example,

1 cu.ft. of air at 60° F., and 14.7 lbs. pressure weighs 0.0764 lb.  
1 cu.ft. of steam at 60° F., and 0.256 lbs. pr. weighs 0.00083 lb.  
1 cu.ft. of mixture has a pressure of 14.956 lbs., weighs 0.07723 lb.

The figures here given for the steam apply to the saturated condition. At the temperature named, no greater weight of steam than 0.00083 lb. could exist in one cubic foot of volume, whether that volume were also occupied with another gas or not, because if there were more than this amount of  $H_2O$  present, its density would be greater than that of saturated steam at 60 degrees and consequently some of the  $H_2O$  would be in the form of water. On the other hand, at this temperature, there may be less than the named amount of steam present. In such a case, the pressure of the steam is less than 0.256 lb. since there is less of it, and therefore it is at a temperature higher than that due to saturation; that is, the steam is then superheated. This is the form in which humidity in the atmos-

phre usually exists. It is also the form in which  $H_2O$  appears in the products of combustion of fuels.

Consider now how the total heat of superheated steam in such a mixture may be determined. Let  $P$  and  $T$  stand for the absolute pressure in pounds per square foot and the absolute temperature in degrees Fahrenheit respectively, of the mixture. These quantities are readily measurable. Also it is possible to figure the weight of one cubic foot of the gases exclusive of the steam, and the weight of steam in one cubic foot of the mixture. Now the pressure of the gases exclusive of steam may be figured from the relation  $P_1 V_1 = RT$ , in which  $P_1$  is the desired pressure,  $V_1$  is the reciprocal of the weight of the gases per cubic foot, and  $R$  may be obtained from works on thermodynamics. (For air,  $R = 53.4$ .) Then the pressure of the steam equals

$$P - P_1.$$

Knowing the pressure of the steam, values of  $h$ ,  $L$ , and saturation temperature,  $t_s$ , may be found from the steam tables; which values, together with the temperature of superheat,  $T$ , make possible the calculation for total heat,  $H$ , according to the expression previously given.

This calculation is laborious, so the following method, sufficiently accurate for engineering purposes, is preferable. The total heat of steam in the form of humidity in air or gases of combustion equals  $1058.7 + 0.455 T_s$ , in which  $T_s$  is the temperature in degrees, Fahrenheit, of the mixture. This is an empirical relation proposed by Professor Diederichs.

In the analyses of boiler, gas engine, and gas producer trials, the heat of the steam in the exhaust gases, counting from room temperature, is required. The heat in the  $H_2O$  before vaporization is  $t - 32$ ,  $t$  being the room temperature. Subtracting this from the expression just given, we have, roughly,

$$1090 + 0.46 T_s - t.$$

**Steam calorimeters** are instruments for measuring the quality of wet steam. Various forms are made depending upon different principles. These depend either upon a transfer of heat by which it can be equated to a measurable quantity or upon the mechanical separation of the entrained moisture. The first is strictly a calorimetric process. Thus if a sample of the steam whose wetness is to be measured is condensed in water, the heat given up to the water is readily measured, and this quantity can be equated to  $h + xL$  in which  $x$  is unknown. The equation then yields the value of  $x$ . Another method is to superheat the steam, in which form its heat content is readily measured.

The mechanical separation method uses an apparatus similar to the ordinary steam separator, and involves weighing both the steam and the separated water.

**Sampling.** As usually arranged only a small sample of the total steam is tested for a quality determination. The accuracy of the result is primarily dependent upon the representativeness of the sample, so that every care should be used to secure a correct one. Unfortunately, it is always uncertain that a reasonably accurate sample has been obtained, but by using the proper shape of sampling pipe, inaccuracy may be reduced to a minimum.

In a horizontal steam pipe, water is apt to run along at the bottom, separated from the steam. If all of this water is passed into the sampling pipe, an undue amount of water appears in the sample. In a vertical pipe moisture runs down the inside of the pipe wall. Undoubtedly some of this should be included in the sample, but it is no easy matter, theoretically or practically, to secure the right proportions. Fig. 63 shows an approved form of sampling tube.



FIG. 63.—Sampling Pipe.

## 30. THE THROTTLING CALORIMETER

**Principles.** Fig. 64 represents the instrument diagrammatically. Steam from a sampling tube enters the steam pipe from which it passes through the orifice  $O$ , about  $\frac{1}{16}$  in. in diameter, into the chamber  $C$  which is open to the atmosphere. The steam is

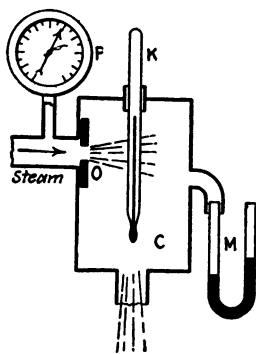


FIG. 64.

Throttling Calorimeter.

throttled upon passing through the orifice, and drops to a pressure only a little in excess of atmospheric. Now the heat contained by 1 lb. of steam at high pressure is greater than the heat contained at low pressure, if superheat does not exist. Upon reaching the chamber,  $C$ , the steam therefore contains an excess of heat, and this excess goes to evaporate the moisture carried in with the steam and to superheat both. In this condition, the heat content is readily measured, which makes possible a heat equation involving one unknown. Before entering the orifice, the heat of 1 lb. of the  $H_2O$  is  $h_1 + xL_1$ . (See page 125 for notation.) In the calorimeter chamber, it is  $h_2 + L_2 + C_p(T_2 - t_2)$ . Neglecting radiation,

$$h_1 + xL_1 = h_2 + L_2 + C_p(T_2 - t_2).$$

The values of  $h$  and  $L$  and  $t_2$  are found from the steam tables, by reference to the pressures indicated by the pressure gage  $P$  and the manometer  $M$ , Fig. 64. A barometer should be read to reduce these pressures to absolute. The temperature  $T_2$  is obtained from the thermometer  $K$ . A sufficiently close value of  $C_p$  is 0.5. These quantities being known, the equation is readily solved for  $x$ .

Certain approximations in the use of the above formula may be made to simplify calculations, which, in view of the uncertainty of sampling, give an ample degree of accuracy. Thus, the pressure in the calorimeter chamber generally is only a few inches of mercury, above atmosphere. If it is taken as 14.7 lbs. absolute,  $h_2 + L_2 = 1150$ , and  $t_2 = 212$ . This obviates the use of the manometer and barometer. Then the equation may be written

$$x = \frac{1150 - h_1}{L_1} + \frac{.5(T_2 - 212)}{L_1}.$$

If the pressure on the entering side fluctuates but little, say 5 or 10 lbs., constant values of  $h_1$  and  $L_1$  may be applied. Now, a value of  $x$  may be calculated for  $T_2 - 212 = 0$ , taking  $h_1$  and  $L_1$  at an average pressure. The increment of  $x$  corresponding to one degree of superheat should then be figured. It is then a simple matter to add to  $x$  (when  $T_2 - 212 = 0$ ) the increment corresponding to any number of degrees of superheat.

For example, when the average pressure is 100 lbs. absolute,  $h_1 = 308$  and  $L_1 = 880$ . With no superheat,  $x = (1150 - h_1) \div L_1 = (1150 - 308) \div 880 = 0.956$ . To this must be added for each degree of superheat an amount equal to  $0.5 \times (1^\circ) \div 880 = 0.00057$ . If the thermometer in the calorimeter chamber shows 222 degrees, the superheat is 10 degrees and the quality of the steam is  $0.956 + 10 \times 0.00057 = 0.962$ , and so on. This method of calculation is convenient for use during an engine or boiler test, since the quality is readily figured from the thermometer reading only, once the constants as obtained above are known.

The errors of the throttling calorimeter are due mainly to false thermometer readings of the superheat, and to radiation of heat from the instrument or fittings. It is best to set the thermometer in the calorimeter without using a well, which may be readily done with a perforated plug of wood or rubber. The thermometer should be moved vertically in the calorimeter

until the hottest part is found when the steam is passing as in regular use. To avoid radiation, all the parts should be well lagged. Not too large an orifice should be used as this raises the back pressure in the calorimeter, and passes more steam than necessary.

(a) **Determination of the Radiation Correction** is sometimes made for extremely accurate work. For this purpose the heat equation should be written

$$h_1 + xL_1 = h_2 + L_2 + C_p(T_2 - t_2) + R$$

in which  $R$  is the heat radiated from the calorimeter from each pound of steam.

Using an apparatus like that shown diagrammatically by Fig. 65, the entering steam may be completely dried. The value of  $x$  is then unity, and the above equation may be solved for  $R$ . The correction  $R$  may then be applied to find more accurate values of the quality when the instrument is in ordinary use. Strictly speaking, the correction applies only to the pressure, room temperature, and orifice of the test. The first two items may vary considerably without much change in the correction.

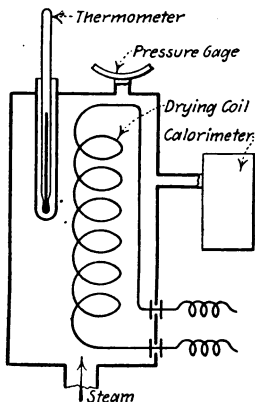


Fig. 65.

A more convenient form for the radiation correction may be had by figuring it as a number of degrees to be added to the thermometer readings.

The proper amount of heat to dry the entering steam is maintained by regulating the current through the drying coils. This may be done by using a water, or other type, rheostat in the line. In order to be sure that the steam is dry, one or two degrees of superheat



may be carried as shown by the thermometer in the drying chamber, its reading being compared with the temperature corresponding to the pressure, as given by the steam tables.

**Problem 30<sub>1</sub>.** Pressure of steam in the calorimeter chamber is 1.5 ins. of mercury. Barometer is 30.8 ins. of mercury. If the steam were saturated in the chamber, what would the temperature be? If the thermometer indicates 280°, how many degrees of superheat are there? What is the total heat of the steam in the calorimeter at 280°? If the pressure of the incoming steam is 84 lbs. gage, what is its quality? *Ans.,  $x = .997$ .*

**Problem 30<sub>2</sub>.** Figure the quality from the data given in Problem 30<sub>1</sub>, assuming the calorimeter pressure to be equal to atmospheric, 14.7 lbs. What is the percentage of error?

**Problem 30<sub>3</sub>.** Figure the quality of steam corresponding to superheats of 25, 35, and 45 degrees, using the approximate method of constants. The steam averages 110 lbs. gage.

**Problem 30<sub>4</sub>.** What is the maximum degree of superheat that it is possible for the steam in the calorimeter to attain if the steam pressure is 95 lbs. gage, and the calorimeter pressure 14.7 lbs. abs.? What is the maximum if the steam pressure is 65 lbs. gage?

**Problem 30<sub>5</sub>.** What is the maximum percentage of moisture that can be shown by a calorimeter under the two conditions named in Problem 30<sub>4</sub>? If the calorimeter pressure is reduced to 10 lbs. abs. by connection with a condenser, what then is the maximum percentage of moisture?

**Problem 30<sub>6</sub>.** If a thermometer reads 3° low on account of radiation, what is the percentage of error resulting from ignoring the radiation correction? Use the data of Problem 30<sub>1</sub>. *Ans., 0.17%.*

## 31. THE SEPARATING CALORIMETER

**Principles.** If the water entrained in steam is mechanically separated, and the weights of the dry steam and separated water are then measured the percentage of water may be readily calculated. Mechanical separation may be effected by deflecting the steam and water through a sharp bend; the water is then thrown out by centrifugal force.

Fig. 66 shows diagrammatically the Carpenter separating calorimeter. The water collects in the calibrated chamber *C* and its amount is measured by the scale *S* placed against a

gage glass. The weight of dry steam is measured by the orifice method according to Napier's rule (see p. 116), the orifice being located at *O*. Since the weight of steam discharged from an orifice into the atmosphere equals a constant times the absolute pressure, an ordinary pressure gage *G* may be calibrated to indicate the weight of steam passing per minute. In operation, a steam sample is passed through the calorimeter, and, after it is thoroughly heated, initial and final readings of the water level are made for a measured interval of time, together with a number of readings of the gage from which an average is calculated. The gage gives the pounds of dry steam passing per minute; multiplying this by the number of minutes gives the total weight of steam. This weight is then divided by the sum of the weights of water and steam to get the quality.

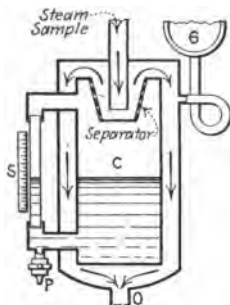


FIG. 66.—Separating Calorimeter.

The chief advantage of the separating over the throttling calorimeter is that it is operative no matter how low the quality.

Experiments have shown that complete separation of the water may be depended upon with a properly designed separator.

(a) **Calibration of the Dry Steam Meter** may be made as for Test 28 (a). A calibration curve for the gage should be plotted.

(b) **Calibration of the Water Gage.** The chamber *C*, Fig. 66, is allowed to fill with water separated from steam passing as in usual operation. A small amount of this water is then drawn off from the pet cock *P* into a flask of cold water (to prevent evaporation). The weights of the flask and contents and the heights of the levels in the water gage before and after the operation are carefully measured. A number of such determinations furnish the data for a calibration curve.

If the water used for the calibration is cold, its weight should be corrected for the difference in density due to the different temperatures. The temperature in usual operation may be taken as that of the steam passing through the calorimeter.

(c) **Determination of Radiation Correction.** Radiation from the instrument may cause a greater amount of water to be thrown down than was in the sample. In the Carpenter instrument, on account of the steam jacket arrangement (see Fig. 66), radiation occurs after the steam has left the separator; the water gage then remains correct, but the orifice measurement may be faulty since the orifice passes wet instead of dry steam. If the gage has been calibrated for the particular conditions of radiation, this calibration takes care of the radiation correction.

If radiation causes an increase of water in the water gage the correction may be determined as follows. Dry steam should be supplied to the calorimeter either with an apparatus as described under Test 30 (a) or by taking the sample from another steam separator. Any water that shows at the water gage of the calorimeter is then due to radiation. If it is figured as so many pounds per minute, the correction in ordinary use is readily applied.

**Problem 31.** If the water gage has been calibrated using the water separated from steam at 150 lbs. gage, what percentage of error will there be when the instrument is used for steam at 50 lbs. gage, the error being due to the change in density of the water between the two temperatures? If the water in the gage glass is 150° F., due to radiation, and the water in the calorimeter chamber is 325° F., will the level in the glass be above or below the inside level, and how much? Will this make material error in the indications, and why?

*Ans.*, 3.6%; 6.6%.

**Problem 32.** If the orifice of the separating calorimeter is used as part of a throttling calorimeter so as to allow for moisture escaping the separator, deduce a formula for the quality of the steam in terms of the weights shown by the separating calorimeter and the quality ( $x_1$ ) as shown by the throttling calorimeter.

## 32. THE CONDENSING CALORIMETER

**Principles.** If a sample of steam the quality of which is to be determined, is condensed either by mixing with or radiating through coils to water, the arrangement constitutes a condensing calorimeter.

Let  $W_s$  = weight of condensate, including moisture in sample,

$h, L, x$  = heat of the liquid, latent heat, and quality of the steam sample,

$W_w$  = weight of condensing water,

$t_w$  = temperature of water before it is heated,

$T_w$  = temperature of water after it is heated,

$T_s$  = temperature of the condensate.

If condensation is accomplished by mixing,  $T_w = T_s$ , otherwise they have different values. Now, if the calorimeter operates continuously, and if radiation is neglected

$$W_s \{ (h + xL) - (T_s - 32) \} = W_w (T_w - t_w).$$

That is, the heat lost by the steam equals that gained by the water.

If the operation is not continuous, the material of the calorimeter absorbs an amount of heat which must be accounted for. This quantity is called the "water equivalent" being the weight of water that absorbs the same amount of heat for a given temperature rise as does the calorimeter. For non-continuous condensing calorimeters, the water equivalent should be added to  $W_w$ .

The equation may be solved for  $x$ , the other quantities being obtained experimentally.  $h$  and  $L$  are found from the steam tables, the pressure of the steam being observed.

A barrel of water on a platform scales makes a non-continuous mixing calorimeter. An injector may be used as a continuous mixing calorimeter. A surface condenser may be arranged as a continuous non-mixing calorimeter.

(a) **Determination of the Water Equivalent.** The condenser should be isolated and filled with water either hotter or colder than the material of the calorimeter. The initial and final temperatures of the water,  $t_1$  and  $t_2$ , and the weight of water  $W$  should be observed. If  $Q$  is the water equivalent, and  $t$  is the initial temperature of the calorimeter, then

$$W(t_1 - t_2) = Q(t_2 - t)$$

since the heat lost by the water equals that gained by the calorimeter, or vice versa. This equation may then be solved for  $Q$ .

The equation neglects radiation. To avoid the consequent error, a second test may be made with the temperatures of calorimeter and water so adjusted that the temperature rise up to room temperature approximately equals that above.

For example, suppose that for the first test water at  $110^\circ$  F. cools to  $80^\circ$ , raising the temperature of the calorimeter from that of the room,  $70^\circ$ , to  $80^\circ$ . The rise of temperature of the calorimeter during the test is  $10^\circ$ . Now, if its initial temperature were  $65^\circ$  and its final,  $75^\circ$ , radiation to and from the room would be balanced. Hence, for the second test, the calorimeter should be previously cooled to  $65^\circ$ , and then heated with water at  $105^\circ$  so that the  $30^\circ$  drop in the water temperature, noted in the first test will bring the final temperature to the desired  $75^\circ$ . The variation will not, of course, be exactly equal above and below the room temperature, but by this method the effect of radiation upon the accuracy of the determination of the water equivalent is made negligible.

The student should note the fact that radiation thus taken care of is an entirely different quantity from that which occurs

in the usual operation of the instrument. For accurate work, the latter should be taken into account as follows.

(b) **Determination of the Radiation Correction.** For non-continuous condensing calorimeters such as the barrel calorimeter, radiation correction may be avoided in actual use in exactly the same way as that described for the water equivalent determination. The flow of steam into the calorimeter is continued sufficiently long to raise the temperature of the cold condensing water as much above room temperature as it was below.

For continuous calorimeters, the radiation correction must be found and applied in the equation, as follows,

$$W_s \{ (h + xL) - (T_s - 32) \} = W_w (T_w - t_w) + R \\ = W_w (T_w + t' - t_w),$$

in which  $R$  is the heat radiated from  $W_w$  lbs. of cooling water and  $t' = R \div W_w$ .

To find  $t'$ , the calorimeter is operated as in usual practice, except that dry steam is furnished to it either by the method of Test 30 (a) or by using a steam separator. The value of  $x$  is then 1, and the equation may be solved for  $t'$ .

The correction applies only to the temperature conditions of the radiation test. If the conditions in ordinary operation are markedly different, the radiation test should be repeated to fit the new conditions. It is possible, however, to keep the initial and final temperatures of the cooling water practically the same for different conditions of the steam samples by varying the rate of cooling water. If this is done, and if the room temperature is not very variable, only a single value of the radiation correction is needed.

**Problem 32.** The data from a barrel calorimeter determination are as follows. Weight of water in barrel before condensing = 350 lbs., temperature = 55° F. Weight after condensing = 370.6 lbs., temperature = 115°. Pressure

of steam sample = 100 lbs. gage. What is the quality, neglecting the water equivalent and radiation? *Ans.*, 0.901.

**Problem 32.** A water equivalent determination for the preceding example shows that 300 lbs. of water will fall from 98.6° to 97.3° F. when placed in the barrel at 60°. What is the water equivalent, and what is the corrected value of the quality? *Ans.*, 10 lbs.; 0.934.

**Problem 32.** The circulating water in a surface condenser rises in temperature from 50° to 86° F., 275 lbs. being used to condense 10 lbs. of a steam sample at 80 lbs. gage. The temperature of the condensate is 98°. What is the quality of the sample? *Ans.*, 0.856.

## FRICTION

Friction is the force which resists the relative motion of two bodies in contact, and is due to the interference of their particles at the surface of contact. The laws controlling this force are different depending upon whether the rubbing materials are fluid or solid. Solid friction may be either from sliding or rolling contact, but in this work sliding friction only will be considered. There is a third case, generally of greatest importance to the mechanical engineer, namely, the friction of lubricated solids. The controlling laws fall between those for solid and fluid friction, resembling the one or the other according to the amount of lubrication.

The laws of fluid friction are well established, but this is unfortunately not true of solid friction. An understanding of the latter is important in the analyses of friction drives, belt forces, etc., and important also to our knowledge of lubricated friction since the rubbing action of solids is a limiting condition of lubricated machine parts.

It has generally been assumed that the friction between two dry solids is a constant fraction of the normal force pressing the solids together, independent of the area of contact and velocity of sliding, and depending only upon the nature of the surfaces. These hypotheses have been discredited, but, thus far, experi-

mentation upon this subject has not proceeded enough to give us more than an idea wherein they are fallacious. It is possible that they are true for certain materials or certain conditions, but unquestionably they are not generally true.

The variation of friction in the general case, depends upon the normal force between the surfaces, the amount of contact area, the velocity of sliding, and the materials of the bodies.

The following table is arranged to give a ready comparison of the several cases. It should be remembered, however, that the statements under solid and lubricated friction are not well established, and are possibly subject to exceptions.

VARIATION OF FRICTION

	Fluid.	Solid.	Lubricated Solid.
Contact area.	Directly proportional.	Independent (approx.)	Directly proportional.
Velocity.	Directly proportional as the square, except at low speeds.	Inversely, at a diminishing rate.	Inversely for low speeds, directly as square root for high speeds.
Character of surface.	Independent.	Varies.	Varies.
Normal force.	Independent.	Directly proportional.	Independent.

It is further known that the frictional resistance to a fluid moving upon a solid is related in some way to the viscosity of the fluid, decreasing as the viscosity decreases. This property has an important bearing upon lubricated solid friction because the viscosity of lubricants varies with temperature, thus introducing another variable in this case.

For solid friction, the working hypothesis is that

$$F = fN$$



in which  $F$  is the force of friction,  $N$  the force normal to the rubbing surfaces, and  $f$  a number less than one called the "coefficient of friction."

### 33. OIL TESTERS

**Principles.** Oil testers are instruments for determining the coefficient of friction in the case of lubricated solids. Their operation generally depends upon the balancing of the friction by a static moment which can be measured; from this and the constants of the instrument, the coefficient may be calculated. Referring to Fig. 67,  $S$  is a revolving shaft to which is fitted a bearing carrying a pendulum. It is arranged that the bearing swing freely and bear on the shaft with a measured force,  $N$ . The oil to be tested is fed to the bearing, and the shaft caused to revolve in a clockwise direction. The force of friction then swings the pendulum to such an angle as to establish equilibrium. If the radius of the shaft is  $r$  ft., then the moment of the frictional force,  $fN$ , is  $fN \times r$ . The weight  $W$  of the pendulum acts on an arm of  $R \sin A$ ,  $R$  being the distance of the center of the weight to the center of the shaft; consequently the moment of the pendulum equals  $W \times R \sin A$ . As this moment equals the moment of the frictional force, we have

$$fNr = WR \sin A,$$

and

$$f = \frac{WR \sin A}{Nr}.$$

For a given set of conditions, all of the quantities on the right

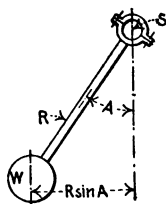


FIG. 67.  
Oil Tester.

of the equation are constant except  $\sin A$  which thus becomes a measure of  $f$ .

The Thurston oil testing machine depends upon this principle. The force  $N$  is applied and varied by means of a spring mounted on the pendulum, by tightening which the two parts of the bearing may be clamped more or less tightly together. The total force,  $N$ , equals twice the tension of the spring plus the weight of the pendulum. There are a pointer and scale on the pendulum by which the total force may be read directly. Another pointer on the pendulum moves over a stationary scale graduated to values of  $WR \sin A \div r$ . A thermometer well is provided in the upper bearing so running temperatures may be ascertained.

The Riehlé oil testing machine carries a hand screw for applying the pressure, and the pressure is measured by means of a scale beam. A second scale beam measures the moment of the turning force induced by friction. By equating this moment to  $fNr$ , as in the Thurston machine, the coefficient of friction may be obtained.

(a) **Constants of the Instrument.** In the Riehlé machine, the only constant generally necessary to test is the value of  $r$ . If desired the indications of the beams may be tested according to the principles of Test No. 1.

With the Thurston machine, it is necessary to find  $WR$ ,  $r$ , and to test the spring.

To find  $WR$ , the pendulum should be swung to a horizontal position and supported on a pedestal resting on a platform scales. A test is then made similar to that for the unbalanced weight of a prony brake, Test No. 6 (a). The result should be multiplied by the horizontal distance between the point of application of the pendulum on the pedestal and the center of motion of the pendulum. This gives the desired moment,  $WR$ .

In addition to measuring the radius of the bearing,  $r$ , its length should be determined, so that the bearing pressure in pounds per square inch of projected area may be calculated.

The graduations on the stationary scale may be tested with these data by calculating their values at any values of the angle  $A$ .

To test the indications of the spring its scale should be determined according to Test No. 2. As the spring is a heavy one, a strength testing machine is required to compress it.

**Problem 33<sub>1</sub>.** Run a series of tests with an oil testing machine to show the relation between coefficient and load in pounds per square inch.

**Problem 33<sub>2</sub>.** Run a series to show the relation between coefficient and velocity.

**Problem 33<sub>3</sub>.** Run a series to show the relation between coefficient and temperature.

### 34. BELT TESTERS

**Principles.** When power is transmitted from one shaft to another by means of a belt running on pulleys, friction is the useful force. Friction sets up tensions in the belt; on one side, a greater amount  $T_1$ , and on the other a lesser,  $T_2$  (see Fig. 68). The difference between these tensions is the net force,  $T$ , which effects a turning effort of the follower. It is shown in works on machine design that

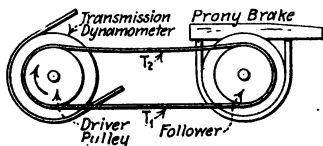


FIG. 68.—Belt Tester.

$$\frac{T_1}{T_2} = 2.72^f$$

in which  $f$  is the coefficient of friction and  $\theta$  is the angle of contact of the belt in radians; the formula being derived on the assumption that  $f$  varies only with the normal force of contact and that centrifugal force on the belt is negligible.

In usual operation, there is always a certain amount of "slip" between the belt and the pulleys. Instead of running at the same linear velocity of points of contact, there is a relative velocity difference between the belt and each pulley rim. This fact invalidates to some extent the formula giving the relation

between the belt tensions, since friction varies with the velocity of rubbing. Furthermore, so little is definitely known about solid friction, that the assumption  $F = fN$  is of doubtful accuracy. It is very likely that, in the case of belts, the coefficient of friction  $f$  varies with the normal force  $N$  as well as with the velocity of slipping. As the formula depends upon an opposite assumption, results of the coefficient of friction obtained from it must be considered of very questionable value. On the other hand we have no better formula, and for this reason it must be used.

Belt testing machines are instruments for testing the efficiency of transmission, for determining the slip under given conditions and for determining the coefficient of friction.

The efficiency of transmission may be found by measuring the power delivered to the belt with a transmission dynamometer and the power delivered to the follower pulley by an absorption dynamometer (see Fig. 68), suitable allowance being made for the friction of the shaft bearings. The difference between these quantities is the losses, which are due to work done by friction in moving through a distance equal to the slip and the work done in flexing the belt, windage neglected.

The slip may be ascertained by counting the R.p.m. of the shafts; the difference between the linear velocities of the pulley rims is the total slip. The slip over either pulley may be found independently by measuring the linear velocity of the belt. This may be done by counting the number of times a chalk mark on the belt appears in a given time or by arranging a piece of metal to project from the edge of the belt so as to trip a revolution counter. Slip should be expressed in per cent of the driving pulley rim speed. Some forms of belt testing machines have a differential gear arrangement by which the per cent of slip may be directly read.

To test for the coefficient of friction, apparatus must be provided for the determination of the belt tensions,  $T_1$  and  $T_2$ , so that the formula may be solved for  $f$ . This is usually done

separate measurements of the sum and difference of the belt tensions; adding these results and dividing by two gives  $T_1$ , knowing which,  $T_2$  is readily obtained.

The turning force of the belt,  $T$ , equals the difference of the tensions. The moment of this turning force equals the torque shown by the friction brake (Fig. 68). Therefore, the difference of belt tensions may be calculated by dividing the torque shown by the brake by the radius of the follower pulley. This radius should be measured to the center line of the belt.

To get the sum of the belt tensions, one type of machine has the driver pulley mounted freely on one arm of a bell-crank lever the other arm of which bears on a platform scales. The belt pull is then indicated on the scales in inverse proportion to the lever arms. Another type of machine has the belt arranged vertically, the follower pulley being freely suspended. The sum of the belt tensions is then equal to the weight of the follower pulley and attachments together with any additional weights used for the purpose of increasing the belt tensions. In both types, the total tension may be varied.

The coefficient varies markedly with the condition of the rubbing surfaces, so every precaution should be used to keep this and other controllable conditions constant. The coefficient also varies with the humidity of the surrounding atmosphere; it is therefore well to make observations of this.

(a) **The Constants of the Instrument** are those of the prony brake (see Test No. 6) and of any lever arms used for getting the tensions. These are readily obtainable by direct measurement.

**Problem 34<sub>1</sub>.** The torque as shown by the friction brake is 100 lb.-ft. If the follower pulley is 2 ft. in diameter, what is the difference between the belt tensions? If the sum of the belt tensions is 200 lbs. and the angle of contact is 190 degrees, what is the coefficient of friction? *Ans.,  $f=0.33$*

**Problem 34<sub>2</sub>.** Describe how a test should be made to show the variation of the coefficient with slip, all other conditions being constant.

**Problem 34<sub>3</sub>.** Describe how a test should be made to show the variation of the coefficient with the normal force of contact, slip and other conditions being constant.

# PART TWO

## THE ANALYSIS OF COMBUSTION

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### THE CONSTITUENTS OF FUEL

THE principal commercial fuels are coals, oils, and gases in their various forms. The elemental constituents of these fuels, which for the most part are common to all of them, are carbon, hydrogen, oxygen, nitrogen, and sulphur. Of these constituents the ones depended upon to yield heat through combustion are carbon and hydrogen. Sulphur has a heat value, but it is an undesirable element in fuel since in coal it goes to form clinker and in gas it may combine with water to make sulphuric acid, these processes accompanying the combustion of the fuel. Nitrogen and carbon in the form of  $\text{CO}_2$  are inert gases and are valueless to combustion.

Carbon and hydrogen occur free or in combination with each other as hydrocarbons or in combination with oxygen. The only combustible carbon-oxygen compound is carbon monoxide which is one of the most important of fuel gas constituents.

Coal may be classified broadly as anthracite, semi-anthracite, and bituminous, distinguishable by the amounts of so-called "volatile matter" they contain. This is burnable material, mainly hydrocarbons, which may be driven from the coal without ignition. The percentages by weight of volatile matter in the three classes of coal when dry are roughly given in the

following table which also shows the amounts that are left of carbon (referred to as "fixed carbon") and ash when the volatile matter is driven off.

### PROXIMATE ANALYSES OF DRY COAL

	Anthracite.	Semi-anthra.	Bituminous.
Volatile matter.....	3 to 10	10 to 20	20 to 35
Volatile matter, average.....	8	17	30
Fixed carbon.....	85	74	60
Ash.....	7	9	10

The ash from coals is the residue when the combustible parts have been burned, and consists of such materials as clay, sand, slate, etc. The ash does not vary with the class of the coal, and may be widely different in proportion and quality in each of the classes. Water is also contained in coals, the amount varying with atmospheric and storage conditions.

It should be observed that anthracite coal is mainly carbon and ash, there being generally very little volatile matter.

A complete chemical analysis of coals (called the "ultimate analysis") is given in the following table which is for representative samples of the American product.

### ULTIMATE ANALYSES OF COALS

	Anthracite.	Semi-anthra.	Bituminous.
Carbon.....	89	82	76
Hydrogen.....	2	4	5
Nitrogen.....	0.8	1	1.5
Oxygen.....	1.2	4	7.5
Ash and sulphur.....	7	9	10

When such an analysis is required it should be obtained from a chemist, as the methods and apparatus are remote from mechanical

experimentation. An analysis yielding the percentages of moisture, volatile matter, fixed carbon and ash, however, may readily be made by the skilled engineer; and often serves all the purposes in the study of combustion. This analysis is called the "proximate analysis."

The kind of coal and its properties as fuel are determined roughly by the locality at which it is mined, but it should be noted that they may vary markedly in samples even from the same mine.

The commercial sizes of coal are as follows, being listed in the order of the size, largest first.

Anthracite	Bituminous
Furnace	Run of mine
Egg	Lump, various sizes
Stove	Nut, various sizes
Chestnut	Screenings, various sizes
Pea	Washed sizes
Buckwheat, No. 1 and 2	

**Oils** used for fuel are composed of different hydrocarbons of the form  $C_nH_{2n+2}$  or  $C_nH_n$  which have a wide range in such physical properties as specific gravity, volatility, burning and flash points. Gasoline and kerosene are the lighter distillates from crude oil.

The percentages by weight of carbon and hydrogen in various fuel oils and their distillates from all parts of the world are not very different. Carbon is generally between 83 and 87 per cent and hydrogen between 11 and 15 per cent, the rest being oxygen, nitrogen and sulphur.

**Gases.** The principal gases used for fuel are illuminating gas, producer gas, natural gas, and blast furnace gas. The following table gives the percentages by volume of the constituents of representative American gases.



## CONSTITUENTS OF GAS FUELS FOR POWER

"Low" means less than 1 per cent.

	Producer.	Illuminating.	Natural.
Carbon monoxide, CO.....	25	15	low
Hydrogen, H <sub>2</sub> .....	12	45	2
Methane, CH <sub>4</sub> .....	2	25	95
Ethylene, C <sub>2</sub> H <sub>4</sub> .....	low	5	low
Benzol, C <sub>6</sub> H <sub>6</sub> .....			
Heavy hydrocarbons.....			
Oxygen, O <sub>2</sub> .....	low	1	low
Sulphureted hydrogen, H <sub>2</sub> S....	low	low	low
Sulphur dioxide, SO <sub>2</sub> .....			
Carbon dioxide, CO <sub>2</sub> .....	5	4	low
Nitrogen, N <sub>2</sub> .....	55	5	2

Blast furnace gas is combustible mainly through CO. It contains a little hydrogen and methane, and is high in nitrogen.

Analysis of these gases must be made by a chemist, with the exception, possibly, of producer gas and blast furnace gas. These may be analyzed by the Orsat apparatus (Test No. 38) if a pipette for the determination of hydrogen is provided, methane being ignored.

## 35. THE PROXIMATE ANALYSIS OF COAL

**Sampling.** When an analysis of a small sample is made to represent a large amount of coal, every precaution should be taken to secure a truly representative sample. For the purposes of boiler and producer tests, this may be done as follows.

As each barrowful or charge of coal is removed from the pile for firing, an amount varying between a handful and a shovelful is withdrawn and put in a closed receptacle to make up a gross sample. The quantity of coal in each of these samplings and of the gross sample necessary for accurate results depends upon

the lump size of the coal, upon its homogeneity, and to a limited extent upon the total amount used. For the smallest sizes of coal of fairly uniform quality, the gross sample may be no more than 100 pounds; under other conditions, it need not exceed 250 lbs. Assuming 200 lbs., the gross sample is broken on a clean surface, preferably a piece of sheet iron, with the assistance of a tamping bar or weight, so that the largest lumps are not more than  $\frac{1}{2}$  in. diameter. This coal is then mixed and piled in the form of a cone. The cone should be "quartered" into four heaps by passing a board through the cone axis in two planes at right angles. Two diagonally opposite heaps are then discarded and the remaining two mixed, broken to  $\frac{1}{4}$  in., coned and again quartered. The process of quartering and discarding should be repeated until about 5 lbs. of coal lumps not larger than  $\frac{1}{4}$  inch are left. Half of this may be put aside in a sealed glass jar for check determinations if desired; the balance is to be run through a coffee mill or other crusher adjusted so that the lumps are reduced to about  $\frac{1}{16}$  in. and less. After this has been thoroughly mixed, a sufficient quantity of it for the proximate analysis is crushed by hand with mortar and pestle until all of that quantity is of such lump size as to pass through a sieve of 20 meshes to the inch. This last amount is used as follows.

(a) **Determination of Moisture, Volatile Matter, fixed Carbon and Ash.** About 5 gms. of the  $\frac{1}{16}$  in. sample are carefully weighed before crushing in the mortar. After crushing, it is spread out on a large watch-glass, care being taken that no fine dust adheres to the mortar or pestle, and placed in a gas oven where it is kept at a temperature of 220° to 230° F. It is then placed in a desiccator to cool so that no moisture may be absorbed from the air. When cool enough to weigh accurately, it is covered, removed from the desiccator and weighed. It is then returned to the oven and left for a half hour, and the cooling and weighing processes repeated. This is continued until the coal shows no further decrease in weight. The difference between the first

weight and the last is the weight of moisture. The percentage is figured on the basis of the weight of the sample including moisture.

The gross sample may lose some of its surface moisture during the processes of quartering and crushing, by evaporation to the air. To avoid this as much as possible, reduction of the gross sample should be performed in a cool room in which the atmosphere is neither very dry nor very moist, and it should be accomplished as quickly as possible.

**To determine volatile matter**, about 1 gm. of the coal is pulverized to pass through a 60-mesh sieve, having been previously weighed. It is then placed in a covered crucible and put in the hottest part of the flame of a Bunsen burner which is 8 ins. in height when burning free. After seven minutes, it is removed, cooled in a desiccator, and weighed. The difference in weight of the sample before and after heating equals the sum of its moisture and volatile matter. Knowing the former, the latter may be readily calculated.

In this determination, it is important to exclude air from the crucible as otherwise some of the fixed carbon may be driven off as  $\text{CO}_2$ . Some experimenters prefer to wet the coal after the first weighing so that the steam generated will displace the air in the crucible before the distillation of the volatile matter.

**Fixed carbon** is found by subjecting the residue from the last test to the heat of a blast flame, the cover of the crucible being removed. Occasional stirring with a platinum wire assists combustion. It is generally a lengthy process to consume all of the fixed carbon; it may be hastened by directing a gentle stream of oxygen from a cylinder filled with that gas into the crucible. When the carbon is all burned, its amount is ascertained by cooling the sample in the desiccator, and weighing as before.

**The ash** is determined by weighing the residue from the last test.

**Sulphur**, which occurs both in the volatile matter and in the ash is sometimes separately analyzed for in the proximate analysis, but its determination is not often necessary.

To shorten the total time necessary for the proximate analysis, it is recommended that the tests for volatile matter, fixed carbon, and ash be made during the time that a separate sample is being dried out for the moisture determination. These tests will show volatile matter and fixed carbon plus moisture; allowance for the latter may be made afterwards.

**(b) Estimation of Hydrogen and Carbon in the Volatile Matter.** For the complete analysis of boiler and producer performance, it is necessary to know the percentage of total

carbon in the fuel and of hydrogen in the volatile matter. As the proximate analysis shows only the fixed carbon, that in the volatile matter may be estimated and added to the amount of fixed carbon to get the total. Professor Lionel S. Marks provides an empirical method of doing this and of estimating the hydrogen in the volatile matter. He has pointed out a relation applying to American coals by means of the curves presented in Fig. 69. The

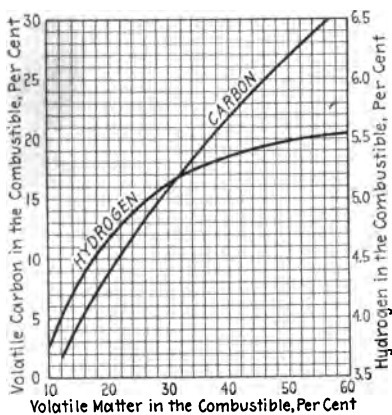


FIG. 69.—Determination of C and H<sub>2</sub> in Volatile Matter.

abscissas of these curves are percentages of volatile matter in the "combustible." By combustible is meant dry coal minus ash. The ordinates are percentages of carbon and hydrogen in the combustible, respectively. All percentages are by weight. Then, given the proximate analysis of a coal,

the percentage of volatile matter shown by it should be divided by the sum of the percentages of volatile matter and fixed carbon to get the percentage of volatile matter in the combustible. Using the curves, the corresponding percentages of hydrogen and carbon may be read. These quantities may then readily be expressed in percentages of the coal as analyzed.

**Problem 35<sub>1</sub>.** The proximate analysis of a coal gives moisture=2 per cent, volatile matter=15, fixed carbon=73, and ash=10. What are the percentages of hydrogen and carbon in the volatile matter, and what is the percentage of total carbon? Base answers on coal as analyzed, not on combustible. *Ans.*, 5.5, 4.0, 78.5%.

**Problem 35<sub>2</sub>.** Give the proximate analysis in the preceding problem on the basis of "dry" coal instead of coal "as received."

*Ans.*, 2.04%, etc.

## THE HEAT VALUE OF FUELS

When used for the generation of power, combustion may be defined as the rapid chemical combination of the oxygen in air with the combustible constituents of fuel, which combination is accompanied by the evolution of heat. It has been pointed out that the elemental combustibles in fuels are carbon, hydrogen, and to a lesser degree, sulphur. The complete combustion of carbon forms carbon dioxide,  $\text{CO}_2$ , and of hydrogen, water,  $\text{H}_2\text{O}$ .

When a unit mass of fuel is burned completely, the heat evolved raises the temperature of the materials entering into the combination and of surrounding objects. If the products of combustion are cooled to the temperature before combustion, then the total heat given up is the "heat" or "calorific value" of the fuel. Or, more briefly, *the heat value of a fuel is the number of heat units that are released by the complete combustion of a unit mass of the fuel.*

It should be noted that to release all of the heat generated, the products must be cooled down to room temperature. This may be done in two ways, namely, so that the  $\text{H}_2\text{O}$  formed by

combustion of the hydrogen remains as steam or so that the  $H_2O$  is condensed. In the former case the latent heat of the steam remains in the products of combustion, the heat released is correspondingly less, and is referred to as the "lower heat value." In the latter case the latent heat of the steam is included in the heat released which is then called the "higher heat value."

Whether the one or the other quantity should be used depends upon the character of the test for which the heat value is needed.

The difference between the two values depends upon the amount of hydrogen in the fuel. For coals, it is comparatively small, but for gases and oils it may be as high as 15 per cent.

The units in which heat values are expressed in the United States are as follows, corresponding French units being used to a limited extent only.

For solids, B.t.u.s per pound.

For liquids, B.t.u.s per pound or per gallon.

For gases, B.t.u.s per standard cubic foot.

**The standard cubic foot of gas** is a cubic foot of gas under standard conditions of pressure and temperature. That this specification is necessary will be seen when it is considered that the mass of gas in a cubic foot depends upon these conditions, and consequently its heat value. The standard conditions of pressure and temperature are either 29.92 ins. of mercury and  $32^\circ F.$ , or 30 ins. of mercury and  $62^\circ F.$  These pressures are the same and equal to 14.7 lbs. per square inch, the difference in the mercury columns being due to the temperature differences.

In this work the 32 degrees standard will be used.

The following gives experimentally determined heat values of the elemental combustibles and of various gases, and of fuels.

#### HEAT VALUES IN B.T.U. PER POUND

Carbon burned to $CO_2$ .....	14,600
Carbon burned to $CO$ .....	4,400
Sulphur burned to $SO_2$ .....	4,000
Hydrogen to $H_2O$ .....	{ 62,000, higher 52,500, lower

## HEAT VALUES IN B.T.U. PER STANDARD CUBIC FOOT (32°)

	Higher.	Lower.
Carbon monoxide, CO.....	342	....
Hydrogen, H <sub>2</sub> .....	346	294
Methane, CH <sub>4</sub> .....	1065	955
Ethylene, C <sub>2</sub> H <sub>4</sub> .....	1680	1560
Benzol, C <sub>6</sub> H <sub>6</sub> .....	4000	3830

## HIGHER HEAT VALUES OF COMMERCIAL FUELS, B.T.U.

Coals.....	11,000 to 15,000 (per lb.)	
Oils.....	18,000 to 20,000	"
Illuminating gas, average.....	650 (per cu. ft.)	
Producer gas . " .....	150	" "
Natural gas " .....	1050	" "

If it is desired to refer the heat values of the gases to the 62 degree standard, it is only necessary to multiply by 0.943, the ratio of absolute temperatures.

Instruments for determining the heat value of fuels are called "fuel calorimeters."

## 36. THE DETERMINATION OF THE HEAT VALUE OF COAL

**Principles.** This may be done by calculation from empirical formulas involving the results of the proximate or ultimate analysis of the coal, or by the use of a fuel calorimeter. The latter method is the more accurate.

Results from coal calorimeters are obtained by burning a small sample of the coal in an air-tight chamber immersed in water. The heat given up to the water and calorimeter parts by the combustion of this coal is measured by the rise of temperature, from which the heat value of the coal may be calculated.

The essential parts of a coal calorimeter are the water vessel, jacketed to prevent radiation, the combustion chamber or "bomb" with a device for igniting the charge, and a ther-

nometer or its equivalent. Ignition is accomplished either by an electric current or by dropping a bit of white hot wire through a tube into the bomb, a valve being arranged to allow its entrance and to prevent the exit of the gases generated. To support combustion, oxygen is charged with the coal, either in the free state from a cylinder under pressure, or in the form of a chemical rich in oxygen.

One of the most convenient of these instruments is the Parr calorimeter with which sodium peroxide is used as a source of oxygen.

**Sampling** of the coal to be tested should be done exactly as for Test 35, except that the last crushing should be to particles that will pass through a 100-mesh sieve. The final sample should be dried in an oven at 220° to 230° F., before it is charged in the calorimeter.

**(a) Determination of Heat Value by the Parr Calorimeter.**

Let  $W$  stand for the weight of water used in pounds,  $w$ , the water equivalent of the bomb, water vessel and parts (that is, the weight of water that would absorb the same amount of heat in the same temperature range), and  $T$ , the temperature rise brought about by the combustion of  $C$  lbs. of coal mixed with the appropriate weight of sodium peroxide. The heat given up by the combustion is then  $(W+w)T$ . Now, 27 per cent of this heat is due to the by-reactions of the sodium peroxide, so that there remains an amount equal to  $0.73(W+w)T$  generated by the coal. Therefore, the heat value of the coal is

$$\frac{.73(W+w)T}{C}.$$

The water equivalent,  $w$ , may be taken as 0.3. Generally 2 liters of water and  $\frac{1}{2}$  gram of coal are used. In English units, these weights are 4.41 and 0.001102 lbs. Combining the constants, we have

$$\text{Heat value} = \frac{.73(4.41+0.3)}{.001102} T = 3117T$$



The temperature  $T$  is the observed temperature minus certain corrections. The hot wire adds some heat which should be allowed for by subtracting  $0.022^{\circ}$  F., from the observed temperature. Anthracite coal is somewhat difficult to burn completely in the bomb, so an additional chemical (about 1 gm.) called "accelerator" is used, the heat from which should be allowed for. The accelerator consists of 2 parts by weight of potassium per sulphate and one of ammonium per sulphate. The corrections are 0.005 degree for each per cent of ash and 0.010 for each per cent of sulphur, to be subtracted.

Radiation from the calorimeter is very small. It may be corrected for roughly by noting the fall in temperature immediately after the maximum temperature has been reached and during an interval of time equal to that between ignition and the occurrence of maximum temperature. The correction should be added to the observed rise.

Radiation may be practically obviated by making the initial temperature of the water about 2 degrees less than that of the room. As the observed rise in temperature is between 4 and 5 degrees, the radiation will be balanced.

The procedure for the test is to add about 9 gms. of sodium peroxide to  $\frac{1}{2}$  gm. of coal, mix thoroughly in the bomb, ignite, and, during the period of heating, to revolve the bomb in the calorimeter so as to circulate the water. This last is accomplished by means of vanes attached to the bomb, the purpose being to quickly make the temperature of the water uniform. The temperature rise is noted from a finely graduated thermometer.

Care should be taken not to allow water to touch the sodium peroxide as they react violently.

(b) **Calculation of Heat Value from the Fuel Analysis.** If the combustibles in coal were in the elemental form, it would be easy to calculate the heating effect from each according to its amount and thus get the heating value of the coal. This, how-

ever, is not so; they are combined as hydrocarbons and otherwise. Now when combustion takes place some heat must be given up to the hydrocarbons to separate the carbon from the hydrogen so that they may recombine with oxygen. That is, some heat becomes latent through dissociation of the elements which lessens the heat available from their combination with oxygen. So there is an interchange of heat in burning coal which makes its heat value dependent upon the previous combination of its elements. We therefore cannot calculate heat values with precision in this way, but certain empirical formulas enable us to make fair estimates. One of them is Mahler's as below.

$$\text{Heat value} = 14,600C + 62,000H - 5400(O + N),$$

the symbols  $C$ ,  $H$ ,  $O$ , and  $N$  standing for the weights in pounds of carbon, hydrogen, oxygen, and nitrogen in 1 lb. of the fuel, respectively. The result is in B.t.u. per pound. This formula uses the ultimate analysis of the coal.

Another formula, using the proximate analysis, is Goutal's. This is

$$\text{Heat value} = 146 \times fc + K \times vm$$

in which  $fc$  and  $vm$  are the percentages of fixed carbon and volatile matter in the fuel as received, respectively, and  $K$  has different values depending upon the amount of volatile matter, as follows:

Volatile Matter, Per cent.	Average $K$ .	Between.
2 to 15	234,	340-220
15 to 30	180,	220-170
30 to 35	171,	170-130

Rough interpolations between these values may be used.

**Problem 36<sub>1</sub>.** A sample of bituminous coal is tested for heat value. Ignition by hot wire. 2.5 liters of water used. Water equivalent of calorimeter = 0.40 lb. Sample weighs 0.6 gm. How much heat is generated and what is the heat value if temperature rise is 4.23° F.? *Ans.*, 14,000 B.t.u.

**Problem 36<sub>2</sub>.** Calculate the heat value of coal giving a proximate analysis as follows: moisture, 2.75; volatile matter, 6.00; fixed carbon, 78.45. Ash, 12.8 per cent. What would be the heat value if the coal were dry?

*Ans.*, 13,200 and 13,600 B.t.u.

### 37. THE DETERMINATION OF THE HEAT VALUE OF GASES AND OILS

**Principles.** The heat values of gases may be calculated readily from their chemical analyses, but this is not true of oils on account of their complex structure. Both oils and gases may be tested for heat value by the use of a properly designed calorimeter. The Junker calorimeter is universally used for this purpose.

This instrument is an ingenious arrangement of heating surfaces surrounded by water, by which all the heat generated by the fuel to be tested is passed to the water. The water is kept flowing through the calorimeter at a constant rate, secured by keeping it under a constant head, and enters continuously at a uniform temperature. Upon leaving the calorimeter, it may be weighed, and the heat added ascertained by noting its rise of temperature. In order that all the heat be given up to the water, the products of combustion must return to the temperature of the fuel and air from which they came, before leaving the calorimeter. This is ascertained by a thermometer placed in the exit gas flue. The water resulting from the burning of hydrogen is condensed and collected, should it be desired to calculate the lower heat value. (See p. 154.)

For gases, a Bunsen burner is used and a small gas meter to measure the fuel in cubic feet. For liquid fuels, a special regenerative burner is used which, together with the lamp con-

taining the fuel, is attached to one arm of a beam balance so that the weight of fuel burned may be measured at any time during the test.

There is no water equivalent of the calorimeter to be considered since, during operation, all parts are at constant temperatures.

Radiation is provided against by air jacketing and polished surfaces, and is negligible.

**Sampling.** If the sample to be tested represents a gas used in a test of several hours' duration, it is best to take a continuous sample covering the whole time of the test as described on page 173, for exhaust gas sampling. This involves rather a large gas container however, and it is more convenient to draw the gas from the main directly into the calorimeter. For this purpose the main should be tapped at a point as near as possible to the place where the gas is used. Then, if a number of heat value determinations are made covering equal intervals of time and separated by equal intervals, their average will be the average heat value of the total gas delivered in the main.

(a) **Determination of Higher Heat Value.** The rates of water and gas flow should be adjusted so that the temperature of the products of combustion upon leaving the calorimeter be that of the room. Then, if  $t$  and  $T$  are the temperatures of the water before and after heating, respectively, in degrees Fahrenheit,  $W$ , its weight in pounds, and  $G$ , the cubic feet of gas burned, we may say

$$\text{Higher heat value} = (T - t)W \div G.$$

In this,  $G$  is the number of cubic feet of gas under standard conditions (see p. 154). To convert the gas meter reading into standard cubic feet, it is necessary to use the relation that the volume of a gas varies directly as its absolute temperature and inversely as its absolute pressure. Consequently, the pressure as well as the temperature of the gas should be noted. This

involves a reading of the barometer and of a manometer set in the gas main.

The heat value determined in this way is not quite the higher value, for the reason that not quite all the water of combustion is condensed. This is because the air entering combustion is not saturated with water vapor (although the fuel may be), but the products of combustion are saturated. The difference in the humidities is due to the water of combustion which is not condensed. To avoid this error, it may be arranged to feed the calorimeter with saturated air and gas. The determinations as made without this precaution are acceptable for most purposes.

**(b) Determination of Lower Heat Value.** This is determined correctly by subtracting from the heat added to the water the amount which came from the vapor in condensing, or its latent heat. To get this amount correctly, we should know the weight of the vapor and its latent heat per pound. The latent heat of steam is generally determined from its pressure by reference to the steam tables. In the case of steam mixed with other gases, the pressure of the steam is only partial (see p. 127) and therefore difficult to determine. Under the conditions existing during the operation of the Junker calorimeter, it is convenient and sufficiently accurate to take the latent heat of the steam as that corresponding to the *temperature* of the exhaust gases, that is, room temperature. Letting  $L$  stand for this heat and  $w$  for the weight of condensation, we have

$$\text{Lower heat value} = \frac{(T-t)W - Lw}{G},$$

the other notation being as used under (a).

An average value of  $L$  may be taken as 1060 for temperatures between 50° and 70° F.

**(c) Calculation of the Heat Value from the Fuel Analysis.** This may be readily and accurately done in the case of gases when

the complete analysis is known. The following rule may be used.

*To calculate the higher or lower heat value of a fuel gas, multiply the higher or lower heat value of each of the constituent gases, in B.t.u. per standard cubic foot, by its volume percentage, as shown by the fuel gas analysis, and divide by 100. Add these results together; the sum is the desired heat value.*

The heat values of various constituents of fuel gases are given on page 155. An example of the calculation is given in columns 1, 2, 6 and 7 of the table on page 257.

**Problem 37<sub>1</sub>.** Figure the higher heat value from a calorimeter test giving data as follows. Pressure of gas = 4 ins. of water. Barometer, 29.4 ins.; temperature of gas, 70° F.; 7.24 lbs. of water raised from 60.7° to 116.3°. Volume of gas burned, 0.875 cu.ft. by meter. *Ans.*, 526 B.t.u.

**Problem 37<sub>2</sub>.** In the preceding problem, if there were .048 lb. of water of condensation, what would be the lower heat value? *Ans.*, 461 B.t.u.

**Problem 37<sub>3</sub>.** Calculate the higher and lower heat values of the producer gas, analysis of which is given in the table on page 149. *Ans.*, 140 B.t.u.

## THE PRODUCTS OF COMBUSTION

The products of combustion of fuels used commercially are called exhaust gases. By their analysis, we learn the completeness of combustion and the directions and amounts of heat losses in the operation of boilers, internal combustion engines, and gas producers.

Commercial combustion, as previously defined (p. 153) involves the chemical combination of the oxygen in air with the combustibles of a fuel, resulting in  $\text{CO}_2$  and  $\text{H}_2\text{O}$ . Air is principally oxygen and nitrogen, the other constituents being negligible in this analysis, and the proportions may be taken as 21 of oxygen to 79 of nitrogen by volume. Now, in order to secure complete combustion, it is necessary always to furnish an excess of air over the amount required to convert all the

carbon to  $\text{CO}_2$  and hydrogen to  $\text{H}_2\text{O}$ . The exhaust gases will therefore contain free oxygen and nitrogen due to this extra air, in addition to  $\text{CO}_2$  and  $\text{H}_2\text{O}$ . Sometimes the combustion is incomplete, in which case the exhaust gases may contain  $\text{CO}$ , free hydrogen, and hydrocarbons. With proper operation, however, these are each less than 1 per cent by volume of the total exhaust gases, and the hydrogen and hydrocarbons are almost always negligible. The experimental determinations are therefore generally only for  $\text{CO}_2$ ,  $\text{CO}$ ,  $\text{O}_2$ , and  $\text{N}_2$ ;  $\text{H}_2\text{O}$  being calculated from the fuel analysis. The analysis of the exhaust gases is found and expressed as percentages by volume.

The ratio, by volume, of the air actually supplied to the fuel to that required for "complete theoretical combustion," is called the "excess coefficient."

**The Principles Involved in the Applications** of exhaust gas analyses depend upon Avogadro's law. It is important that this law be thoroughly understood. Following is a statement of it.

*In a given volume of any gas, there is always the same number of molecules, regardless of the nature of the gas, provided the conditions of pressure and temperature are constant.*

We therefore know that in a vessel containing a mixture of gases, there is the same number of molecules as though only one of them or of any gas existed there alone, irrespective of the number of atoms in the molecules. Also, the number of molecules in 2 cu.ft., for example, is twice the number in one; that is, the number of molecules in a gas is directly proportional to its volume.

It follows from Avogadro's law that, for given conditions of pressure and temperature, the volume occupied by a single molecule of any gas is a constant quantity. Since all of the determinations from exhaust gas analysis are relative, it will be convenient to conceive this volume as a cubic foot and to use the unit, "molecule-foot," to express it.

Elemental gases are supposed to have two atoms to a molecule. For that reason they are written  $O_2$ ,  $N_2$ ,  $H_2$ , etc. Carbon does not occur as a gas except in combination with other elements but it will be convenient to consider hypothetical gaseous carbon having a single atom to the molecule.

The relative weights of the elements under consideration, compared with that of hydrogen (that is, their atomic weights), are as follows.

$$H=1, \quad O=16, \quad N=14, \quad C=12$$

The molecular weight of a gas equals the sum of the weights of the atoms in its molecule. Thus,  $H_2$  weighs  $1+1=2$ ,  $O_2$  weighs  $16+16=32$ ,  $CO_2$  weighs  $12+16+16=44$ , and so forth.

As an illustration of the way in which these principles are applied, consider a mixture of gases consisting of 10 per cent of

		Nitrogen			
	⊙	⊙	⊙	⊙	⊙
	Carbon Dioxide				
⊙	⊙	⊙	⊙	⊙	⊙
∞	∞	∞	∞	∞	∞
	Oxygen				
⊙	∞	∞	∞	∞	∞
Carbon Monoxide					

FIG. 70.

Molecule-feet.

$CO_2$ , 1 of  $CO$ , 9 of  $O_2$ , and 80 of  $N_2$ ; all per cents being by volume. Since these gases are expressed relatively (the basis of the comparison being 100 parts, i.e., per cent) we may consider their percentages as cubic feet, or as molecule-feet. Fig. 70 represents the per cents of the gases considered thus, each per cent is a cubic foot and each cubic foot contains a single molecule.

Suppose we wish to find the ratio by weight of the oxygen in the gas to the carbon. There are 9 molecule-feet of oxygen as such, 10 molecule-feet of oxygen in the form of  $CO_2$ , and one-half a molecule-foot as  $CO$ , making a total of 19.5 molecule-feet. Since the molecular weight of  $O_2$  is 32, the total weight of  $O_2$  is  $19.5 \times 32$ . Similarly, the number of molecule-feet of carbon is  $9+1=10$ , having a weight of  $10 \times 12$ . Dividing the weight of oxygen by the weight of carbon,  $19.5 \times 32 \div 120$ , we get the



desired ratio of weights, 5.2. Since this is a ratio, it may be expressed in any units of weight, for example, 5.2 lbs. of oxygen per pound of carbon. In this way we may figure the weight in pounds of any or all of the gases or of the elements composing them per pound of any of the constituents.

The total carbon used in combustion may be measured when in the form of fuel. If this weight is multiplied by that of any of the products of combustion, expressed in pounds per pound of carbon, the result is the total weight of that product of combustion. This follows from the assumption that all of the carbon in the fuel appears in the exhaust gas analysis.

### Theoretical Reactions of Fuels

**Products from Coal Combustion.** In the burning of pure carbon, if complete combustion could take place with no excess of air, the reaction would be



showing that the exhaust gas would consist of 21 per cent of  $\text{CO}_2$  and 79 per cent of  $\text{N}_2$ . Now, if more than this amount of air is used, the sum of the free oxygen and of the  $\text{CO}_2$  in the exhaust gas will bear the same relation by volume to the nitrogen as the oxygen did in the air, since the volume of the oxygen is not increased by combination with carbon. Consequently the sum of  $\text{O}_2$  and  $\text{CO}_2$  is 21 per cent, and  $\text{N}_2$  is again 79 per cent.

Ash does not vary this relation, but the effect of hydrogen, oxygen, and nitrogen in the coal is to alter it. The last two have slight effect, but the hydrogen needs oxygen from the air to burn to water, and as the water does not appear in the exhaust gas analysis, a corresponding amount of oxygen disappears. This increases the percentage of nitrogen. The formation of  $\text{CO}$  instead of  $\text{CO}_2$  has an opposite effect, since the oxygen entering

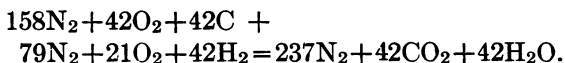
the combination makes twice the volume of CO; on the other hand CO is generally low.

Anthracite and semi-anthracite coals, being low in hydrogen, will yield exhaust gas having between 79 and 80 per cent of N<sub>2</sub>. Bituminous coals will yield between 80 and 81.5 per cent. Results outside of these figures are questionable.

It is obvious that the less air is used, the less will be the free oxygen in the exhaust gas, and the greater will be the CO<sub>2</sub>. It will be shown (Test No. 50) that the less the air used the more economical is the combustion, provided that there is not so little air as to cause material loss from incomplete combustion. Consequently high CO<sub>2</sub> is a desirable indication.

Fig. 71 shows the variation in the products of combustion of carbon and of anthracite coal, for various amounts of air, assuming complete combustion.

**Products of Oil Combustion.** Since the proportions of the elemental constituents of fuel oils are approximately constant for all of them, the products of their complete combustion will also be constant under the same conditions of air supply. An average analysis is C, 84, and H<sub>2</sub>, 14 per cent by weight. The ratio of volumes of carbon to hydrogen in molecule-feet is  $84 \div 12$  to  $14 \div 2$  or 1 to 1. As it takes 100 molecule-feet of air to burn 21 of carbon to CO<sub>2</sub>, and half that much to burn an equal amount of hydrogen to H<sub>2</sub>O, the following will be the reaction assuming no excess air and complete combustion.



As the H<sub>2</sub>O disappears, the total volume of the products of combustion is  $237 + 42$  molecule-feet. This divided into 42 gives the percentage of CO<sub>2</sub>, about 15 per cent. This is the maximum value in the exhaust gas from usual oil fuels.

Fig. 72 shows the variation in the exhaust gas from oils for

various amounts of air supplied. It should be noted that in the case of internal combustion engines, oil or oil vapor has a limited range of explosibility so that the excess of air cannot be very high. If the proportions of the exhaust gases are markedly different from those shown by Fig. 72, the  $\text{CO}_2$  being low, it is probable that incomplete combustion is taking place and that hydrocarbons are appearing as  $\text{N}_2$ .

**Products of Gas Combustion.** The proportions of hydrogen and carbon in the different fuel gases vary widely and this causes

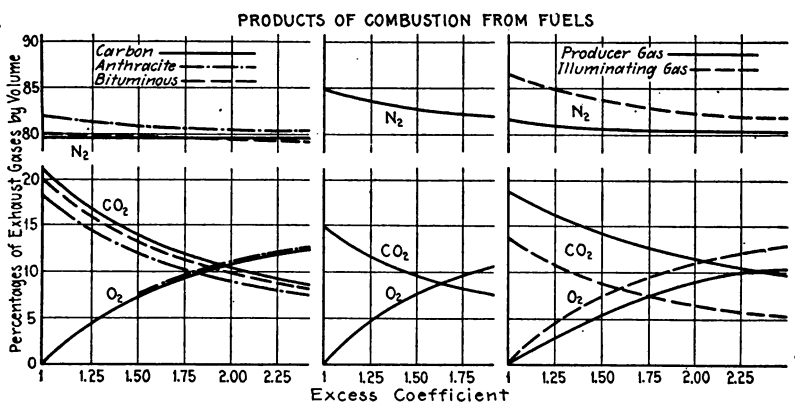


FIG. 71.

FIG. 72.

FIG. 73.

wide variations in the exhaust gases. Furthermore, some of the fuel gases contain high percentages of nitrogen, which makes that gas correspondingly high in the exhaust, and the  $\text{CO}_2$  low. Knowing the analysis of a given fuel gas, the number of molecule-feet of  $\text{C}$ ,  $\text{H}_2$ ,  $\text{O}_2$ , and  $\text{N}_2$  may be readily counted, from which the theoretical reaction may be figured as for oil. A convenient method of counting the molecule-feet is shown in the table, p. 257, under columns 3, 4, and 5.

Fig. 73 shows the variation in the products of combustion

of representative producer gas and of illuminating gas, for various amounts of air, assuming complete combustion.

**Calculation of the Exhaust Gas Analysis to be Expected from a Given Fuel.** It is often advisable to make such calculations in order to check the actual analysis obtained by experiment.

Let  $c$ ,  $h$ ,  $g$ , and  $n$  be the number of molecule-feet of  $C$ ,  $H_2$ ,  $O_2$ , and  $N_2$ , respectively, in 100 molecule-feet of the fuel. The analysis of solid and liquid fuels gives the percentages of these elements by weight; dividing the percentages by the molecular weights of the elements, respectively, gives the number of molecule-feet. Analyses of gases give percentages by volume; the molecule-feet may be counted as in columns 3, 4, and 5 of the table on page 257. Let  $X$  be the ratio by volume of the air supplied in the actual combustion to the air needed as shown by the theoretical reaction, that is, the excess coefficient.

Since there are  $c$  molecule-feet of carbon in the fuel, there will be  $c$  molecule-feet of  $CO_2$  in the exhaust. Since carbon requires 1 molecule-foot of  $O_2$ , and hydrogen half as much, for complete combustion, the amount of oxygen needed is  $c+0.5h$ . But the fuel furnishes  $g$  molecule-feet of  $O_2$ . There will then be  $(X-1)(c+0.5h-g)$  molecule-feet of oxygen as excess which will appear as free oxygen in the exhaust gas analysis. The nitrogen accompanying the oxygen supplied as air is  $\frac{79}{21}X(c+0.5h-g)$ . In addition to this nitrogen, there will appear in the exhaust  $n$  molecule-feet of nitrogen from the fuel. Adding all these volumes gives the total volume of the products of combustion in molecule-feet, which, simplified, is

$$c + (4.75X - 1)(c + 0.5h - g) + n.$$

Calling this  $T$ , we have for the volume percentages of the exhaust gases,

$$\%CO_2 = 100c \div T$$

$$\%O_2 = 100 \{ (X-1)(c+0.5h-g) \} \div T$$

$$\%N_2 = 100 \{ 3.75X(c+0.5h-g) + n \} \div T$$

Nitrogen may also be obtained by subtracting the percentages of  $\text{CO}_2$  and  $\text{O}_2$  from 100.

As an example in the use of these formulas, consider the combustion of the producer gas the analysis of which is given on page 149, assuming the excess coefficient to be 1.5. Counting the molecule-feet, we have

$$\begin{aligned} c &= 25 + 2 + 5 = 32 \\ h &= 12 + 4 = 16 \\ g &= 12.5 + 5 = 17.5 \\ n &= 55 \end{aligned}$$

$$\begin{aligned} \text{from which, } T &= 32 + (4.75 \times 1.5 - 1)(32 + 0.5 \times 16 - 17.5) + 55 \\ &= 225 \end{aligned}$$

$$\begin{aligned} \text{and } \% \text{CO}_2 &= 100 \times 32 \div 225 = 14.2 \\ \% \text{O}_2 &= 100 \{ (1.5 - 1)(32 + 0.5 \times 16 - 17.5) \} \div 225 \\ &= 5.0 \\ \% \text{N}_2 &= 100 - 14.2 - 5.0 = 80.8 \end{aligned}$$

It should be remembered that these percentages are what would result under complete combustion; the actual percentages will be somewhat different if this condition does not obtain, as would be indicated by the appearance of  $\text{CO}$  in the exhaust.

It is convenient to express the corresponding formulas for coals and oils in terms of their constituents by weight, since the fuel analyses are so expressed. Let  $C$  and  $H$  stand for the weights of carbon and hydrogen per pound of fuel, respectively. The other constituents may generally be neglected, since in the fuels considered they have but small effect upon the exhaust gases. Now  $C \div 12$  and  $H \div 2$  are proportional to  $c$  and  $h$ , respectively. Substituting these values for  $c$  and  $h$  in the equa-

tions for gas fuel combustion, and neglecting  $n$  and  $g$ , we have, for coals and oils,

$$\%CO_2 = \frac{100 C}{C + (4.75X - 1)(C + 3H)}$$

$$\%O_2 = \frac{100 (X - 1)(C + 3H)}{C + (4.75X - 1)(C + 3H)}$$

$$\%N_2 = 100 - \%CO_2 - \%O_2$$

If the analysis of a fuel to be used during a test is approximately known, it is often of value to calculate the exhaust gas analysis to be expected from it using different amounts of air, and to plot the results as curves similar to Figs. 71, 72, and 73. By comparing the actual results of the analyses during the test with those shown by the curves, their correctness may be roughly checked; and, when hydrocarbons appear as nitrogen, their presence may be deduced.

**Air Required for Complete Theoretical Combustion, per Pound of Fuel.** This is the amount that would be used if complete combustion could occur without excess of air. **For coals and oils,** it is convenient to express it in pounds of air per pound of the dry fuel.

A molecule of oxygen, weighing 32, unites with 1 of carbon, weighing 12, to make  $CO_2$ . Consequently there will be  $\frac{32}{12}$  lbs. of  $O_2$  used for every pound of C burned. As oxygen is 23 per cent by weight of the air containing it, the air required to burn each pound of C is  $\frac{32}{12 \times .23}$  lbs., or

11.6 lbs. of air to burn 1 lb. of C.

Two molecules of hydrogen, weighing 4, unite with 1 of  $O_2$  to make  $H_2O$ . Consequently there will be  $\frac{32}{4} \div .23$  lbs. of air required to burn each pound of  $H_2$ , or

34.8 lbs. of air to burn 1 lb. of  $H_2$ .

If  $C$  and  $H$  stand for the weight of carbon and hydrogen in one pound of the fuel, and if the fuel contains no oxygen by which combustion may be supported, then the total amount of air needed per pound of fuel is  $11.6C + 34.8H$ . If the fuel contains  $O$  lbs. of oxygen per lb., a corresponding amount less of air is needed, namely  $\frac{100}{23}O = 4.35 O$ . In the general case, then,

$$\text{Air required} = 11.6C + 34.8H - 4.35 O \text{ lbs. per lb. fuel.}$$

**For Gas Fuels,** the air required is generally expressed in cubic feet of air per cubic foot of fuel, the conditions of temperature and pressure being the same for both.

Letting  $c$ ,  $h$ , and  $g$ , stand for the number of molecule-feet of  $C$ ,  $H_2$ , and  $O_2$ , in 100 molecule-feet of the fuel, counted as in columns 3, 4, and 5 of the table on page 257, there will be required  $c$  molecule-feet of  $O_2$  to burn the  $C$  to  $CO_2$ , and  $0.5 h$  molecule-feet of  $O_2$  to burn the  $H_2$  to  $H_2O$ . Since there are  $g$  molecule-feet of  $O_2$  contained by the fuel, only  $(c + 0.5h - g)$  molecule-feet of  $O_2$  will be required from the air to burn 100 molecule-feet of the fuel. Therefore,

$$\text{Air required} = \frac{c + 0.5h - g}{100 \times .21} \text{ cu. ft. per cu. ft. fuel.}$$

### 38. THE ANALYSIS OF EXHAUST GAS

**Principles.** The gas analysis apparatus depends upon the separate absorption of the products of combustion by certain reagents. A measured volume of the gas is brought into contact with one of these reagents which removes the  $CO_2$ , but does not take up any of the other constituents. The  $CO_2$  is then determined by measuring the diminution in volume. In this way the volumes of  $CO$  and  $O_2$  may also be found. The residue is

assumed to be  $N_2$ . It is convenient to make the original volume 100 units, as cubic centimeters, in which case the differences are per cents.

The Orsat apparatus in its various forms is in general use for engineering analyses of exhaust gas. It consists of a measuring flask,  $B'$ , called the burette (see Fig. 75), a distributing tube,  $T$ , and three so-called pipettes,  $DD'$ ,  $OO'$ , and  $MM'$ , containing the reagents. The gas sample is displaced from the burette into each pipette in turn by filling the burette with water from

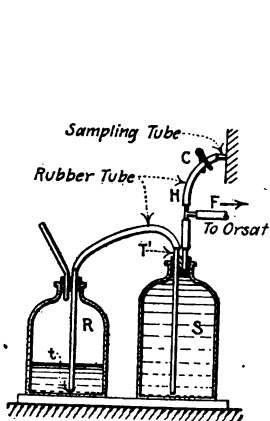


FIG. 74.—Gas Sampling Bottle.

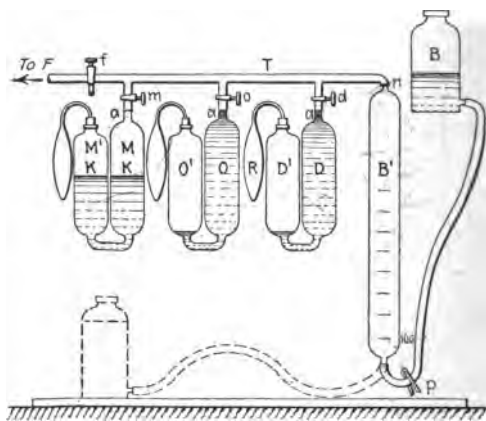


FIG. 75.—Orsat Apparatus.

the bottle  $B$ . The right leg of the pipette is completely filled with reagent just before the gas enters; the gas displaces it into the left leg. Rubber bags,  $R$ , seal the left legs of the pipettes so that the reagents will not deteriorate through contact with the atmosphere.

If desired, the apparatus may be provided with a fourth pipette for determining hydrogen.

There are many other types of exhaust gas analysis apparatus,



the most important of which are recording instruments giving the percentage of  $\text{CO}_2$  in boiler flue gas continuously.

**Reagents.** For  $\text{CO}_2$ , a 1 to 2, by weight, solution of potassium hydroxide (KOH) in water. Enough for charging one pipette, about 150 cc., may be made by dissolving 60 gms. of KOH with 120 gms. of water. The reagent will absorb 40 times its own volume of  $\text{CO}_2$ .

For  $\text{O}_2$ , the reagent is a mixture of water, potassium hydroxide, and pyrogalllic acid in the proportions of 20 to 24 to 1 by weight. The potassium hydroxide should first be dissolved in the water and then the pyro, in powder form, added. Enough for one pipette charge may be made by using 83 gms. of water to 100 of KOH to 4.25 pyro. The absorptive capacity is twice its own volume.

For CO, the reagent is a 3 to 40 solution of cuprous chloride ( $\text{CuCl}$ ) in water. To this should be added just enough strong ammonia to cause a barely perceptible odor. Enough for one pipette charge may be made by using 11 gms. of cuprous chloride to 147 gms. of water. The absorptive capacity is four times its own volume.

**Sampling.** For use in the apparatus, a small amount of the total gas is led from the main body through a sampling tube. The proper construction of this tube depends upon whether the gas is flowing in a large flue, as in boiler work, or in a comparatively small pipe, as in gas engine work. For the latter it is sufficient to tap into the exhaust pipe a pipe of about  $\frac{1}{4}$ -inch diameter. In a boiler flue, however, the composition of the gas may vary through any section of the current, so means should be provided for drawing gas from various parts of the section. One method of doing this is to run the sampling pipe diagonally across the flue, the end of the pipe being sealed and holes being drilled across the length in the flue. The holes should increase uniformly in size from the point of entrance of the sampling pipe to its end, as otherwise more gas will be drawn from the

holes in the near end of the pipe. Another method, advocated by the A.S.M.E., is to use a sampling box as shown in Fig. 76. The pipe should enter as nearly as possible to the furnace, but beyond the combustion chamber or last pass in which combustion is taking place. This should be done because the farther the gas gets from the furnace, the greater is its opportunity to become diluted with air entering through leaks in the brick work or setting.

The sample may be collected with an apparatus such as shown by Fig. 74, consisting of two 3- to 5-gallon flasks connected as shown. The flask *S* is used to receive the sample. To start with, it is filled with water. During collection, the water is syphoned into flask *R* through *T'* and the gas flows through *H* into the space evacuated, branch *F* being closed.

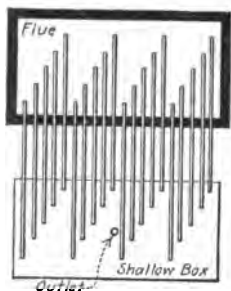


FIG. 76.—Flue Gas Sampling Box.

All the air in the tube *H* should be displaced before taking the sample. This may be done either by raising the flask *R* above the level of the sampling tube so that water will syphon into the tube *H*, or by drawing a small charge of gas into flask *S* in the regular manner and

then discharging it through a three-way cock at the end of the branch *F*. The latter procedure leaves flue gas in the tube *H* very slightly diluted with air.

*The sample should be collected at a uniform rate, and, if more than one sample of the same gas or for the same test is to be analyzed, time periods during which collections are made should be equal in order to get a correct average result. The rate may be kept practically uniform by placing flask *R* several feet below *S* and by choking the end of the syphon at *t* with a sliver of wood. The change in the rate due to the decreasing distance between the water levels in the flasks may then be inconsiderable.*

For boiler and gas engine tests, it is well to collect the sample during half hour or hour periods, and during the collection of each sample to analyze the preceding one.<sup>1</sup> This has the advantage of showing the uniformity of combustion as the test proceeds. It is also well to have a duplicate sampling outfit, collecting gas at a very slow rate so that the sample will cover the whole test. This may be analyzed at the convenience of the experimenter; the results should check the average of the separate analyses.

(a) **Determination of  $\text{CO}_2$ ,  $\text{O}_2$ ,  $\text{CO}$  and  $\text{N}_2$ .** The Orsat apparatus is first connected with a rubber tube to the branch *F* leading from the sample bottle (see Figs. 74 and 75). The reagents in the pipettes are brought to the levels *a*, by manipulation of the bottle *B*, the three-way cock *f* being closed. This being done, and the cocks, *d*, *o*, and *m*, closed, the water level in the burette is raised to the point *n*, thus expelling the air previously contained through the three-way cock *f* which is open to the atmosphere for that purpose. The pinch-cock *C* on the sampling bottle flue connection is now closed, and the flask *R* raised so as to put the sample under pressure. The cocks between the sample bottle and the burette are then opened, and the bottle *B* lowered so that part of the sample will flow into the burette. The gas now in the burette is diluted with air or other gas which was in the distributing tube and in the rubber tube leading to the sampling bottle. To get a purer charge, this first one is discarded through the three-way cock, *f*, and another one brought into the burette, in the same way as the first, which may be accepted for analysis. This charge should be somewhat greater than 100 cc.

Having secured this sample by closing the three-way cock, the bottle *B* is raised a trifle until the water level in the burette reaches the lowest graduation which marks 100 cc. of gas. As the sample has been compressed by this rise in water level, the pressure should be made equal to atmospheric by opening the

three-way cock for an instant, the pinch cock *p* being closed so as to maintain the water level at the lowest graduation.

The sample should now be worked back and forth in the KOH solution to remove the  $\text{CO}_2$ . After this has been done for a few minutes, it should be returned to the burette, care being taken that the reagent is at its original level, *a*; the cock *d* should be closed; and a volume measurement of the gas made. This must be done with the sample under the same condition of pressure as when originally measured, that is, atmospheric. For this reason, the water levels in the burette and in the bottle *B* should be coincident when the reading is made. Before making the reading it is well to allow the walls of the burette to drain for a minute. After the reading the gas is returned to the KOH solution and worked a few more times, then measured again. If the volume is the same the result is satisfactory, if not, the process should be repeated until all the  $\text{CO}_2$  has been removed as indicated by constant volume. The diminution of volume is the per cent  $\text{CO}_2$ .

The  $\text{O}_2$  and CO are removed and measured in the same way and in the order named. The difference between the sum of the per cents of  $\text{CO}_2$ ,  $\text{O}_2$ , and CO, and 100 may be taken as the percentage of  $\text{N}_2$ .

**Precautions in Operating.** Before using the apparatus, all connecting tubes and cocks should be tested for leaks by putting air in the system under pressure or partial vacuum with the bottle *B*, and noting whether the volume remains constant as shown by the level in the burette.

Analyses should be made at a uniform temperature of not less than  $60^\circ \text{F}$ . During operation, the apparatus should not be exposed to a changing temperature, or placed where drafts or sun may strike it. The corresponding changes in the volume of the sample would give false results.

Water absorbs all of the gases to some extent. To avoid the resulting errors, it is well to use water that has been previously

saturated with exhaust gas, so that it will not take up any more. To do this the exhaust gas is caused to bubble through the water used in the sampling flask and in the burette.

If a little coloring matter, such as red ink, is added to the water used in the burette, it will aid in the reading of the graduations.

It is well to keep a record of the volumes absorbed by each reagent so that it will be known when their absorptive capacity has been reached. The volume of reagent in a pipette is about 150 cc.

**(b) Calculations from Exhaust Gas Analyses.** The principles upon which these depend are stated and illustrated on pages 162 to 165. They should be thoroughly understood.

The results for coal combustion are generally based upon 1 lb. of dry coal, while those for fuel gas are based on one standard cubic foot of the fuel. Therefore two different sets of formulas will be deduced for the two cases.

It is convenient to apply the following rule for a number of purposes. *The density of a gas, in pounds per standard cubic foot, equals the density of hydrogen times the ratio of molecular weights of the gas and hydrogen.* The density of hydrogen is 0.00559 lb. per standard cubic foot; consequently the density of gaseous carbon, for instance, is  $0.00559 \times \frac{12}{2} = 0.0335$ .

The following notation will be used for the two cases in common

$D, M, O, N, H$  = percentages of carbon dioxide, monoxide, oxygen, nitrogen, and hydrogen, by volume, in the exhaust gas, respectively.

$V_a$  = volume of dry exhaust gas in cubic feet per pound of coal or per cubic foot of fuel gas.

$W_a$  = weight of air supplied in pounds per pound of coal, or per cubic foot of fuel gas.

$W_d$  = weight of dry exhaust gas in pounds per pound of coal, or per cubic foot of fuel gas.

$W_o$  = weight of  $H_2O$  in the exhaust gas, per pound of coal, or per cubic foot of fuel gas.

$X$  = excess coefficient; that is, the ratio of the amount of air supplied to that required for complete theoretical combustion.

It should be borne in mind that the weights here tabulated are quantities resulting from the *combustion of 1 lb. of dry coal*, or from 1 *standard cubic foot of fuel gas* (see page 154).

**Formulas for Coal Combustion.** In 100 molecule-feet of the exhaust gas, there are  $D+M$  molecule-feet of carbon, since each per cent of  $CO_2$  and  $CO$  contains 1 molecule-foot of carbon. Therefore, the ratio  $100 \div (D+M)$  is the number of cubic feet of dry exhaust gas per cubic foot of gaseous carbon contained by it. This is for *dry* exhaust gas (that is, the volume of the water actually contained is omitted) because the analysis does not take account of the  $H_2O$ . If it is divided by the weight of a cubic foot of gaseous carbon, 0.0335 lbs., the result is the number of cubic feet of dry exhaust gas per pound of carbon in it, or  $2980 \div (D+M)$ . Multiplying this by the weight of carbon that is gasified from 1 lb. of dry coal, we have the volume of dry exhaust gas per pound of dry coal, or

$$V_a = 2980 \frac{C_g}{D+M},$$

in which  $C_g$  is the weight of carbon gasified. This quantity is all of the carbon in 1 lb. of dry coal,  $C_t$ , except that lost through the grate and removed with the ash,  $C_a$ ; or

$$C_g = C_t - C_a$$

assuming carbon deposited as soot to be negligible.  $C_t$  may be estimated from the proximate analysis and  $C_a$  measured from the

total ash (Test 50 (c)). If  $C_i$  is found from the analysis in terms of percentage of the coal as analyzed, that is, wet coal, it should be divided by 100 minus the percentage of moisture to reduce to weight per pound of dry coal.

To find the weight of air supplied per pound of dry coal, we may assume that all of the nitrogen in the exhaust gas comes from the air, a very nearly correct assumption since coal contains but little nitrogen. Now, there are  $28 N$  weights of  $N_2$  in the exhaust gas, and  $12(D+M)$  weights of C; 28 and 12 being the molecular weights, respectively. Therefore, the weight of  $N_2$  supplied by the air to each pound of carbon gasified is  $\frac{28N}{12(D+M)}$  pounds. Dividing this by the proportion of  $N_2$  in air, 0.77 by weight, gives the pounds of air supplied to a pound of the carbon in the exhaust gas. Multiplying by the weight of carbon gasified from 1 lb. of dry coal gives the weight of air supplied per pound of dry coal, or

$$\begin{aligned} W_a &= \frac{28NC_g}{12 \times .77 \times (D+M)} \\ &= 3.04 \frac{NC_g}{D+M} \end{aligned}$$

The weight of dry exhaust gas is that of the air supplied plus that of the carbon combined with it, approximately, or

$$W_g = W_a + C_g$$

This is approximate since some of the oxygen in the air has combined with the  $H_2$  in the coal to form water which is separately counted. This, however, is comparatively small. Also other constituents than carbon from the coal are in the dry exhaust gas, oxygen and sulphur, for instance. With bituminous

coals, the oxygen may be as high as 10 per cent by weight, but this amount would figure as less than 1 per cent of the dry products of combustion.

Water vapor in the exhaust gas comes from three sources, namely from combustion of hydrogen, from moisture in the coal, and from the moisture in the air supplied. The first two only need be considered.

If  $H_t$  is the weight of hydrogen in 1 lb. of dry coal, the resulting water vapor, if all the hydrogen is burned, will be  $9 H_t$ , since the ratio of weights of  $H_2O$  to the  $H_2$  in it is  $(2+16) \div 2 = 9$ . If  $m$  is the weight of moisture in the coal per pound of dry coal, then the total weight of  $H_2O$  in the exhaust gas, per pound of dry coal is

$$W_w = 9H_t + m$$

The assumption that all the  $H_2$  is burned is sufficiently accurate under usual operating conditions.

If  $H_t$  and  $m$  are found from the proximate analysis (see Test 35, (a) and (b)) as percentages of the coal as analyzed, they should be divided by 100 minus the percentage of moisture to reduce to pounds per pound of dry coal.

The weight of carbon incompletely burned, per pound of dry coal, may be calculated from the CO appearing in the exhaust under the assumption that there is no incomplete combustion through unburned hydrocarbons. The method is the same as for the determination of the weight of air. Let  $W_c$  be the desired weight. Then

$$\begin{aligned} W_c &= \frac{12MC_g}{12(D+M)} \\ &= \frac{MC_g}{D+M} \end{aligned}$$

To find the excess coefficient, it is necessary to divide the



value of  $W_a$ , as deduced above, by the expression for the air required as given on page 171, then, neglecting  $O_2$  in the coal,

$$X = W_a \div (11.6C_i + 34.8H_i).$$

The formulas above also apply to the combustion of oils if  $C_i$  is substituted for  $C_g$ , and if  $m$  is left out of the expression for  $W_a$ .

**Formulas for Gas Combustion.** These are deduced upon the assumption that all the carbon in the fuel appears in the exhaust gas analysis, and that none other does. Contrary to this, in the case of internal combustion engines, is the fact that some carbon is left as a deposit in the cylinder, but it is so small compared with the total carbon used as to be negligible. Also, if the lubricating oil burns, it will appear as  $CO_2$  in the exhaust, so care should be taken to avoid this condition. Hydrocarbons in the exhaust, too, will prevent the correct application of the formulas. If the CO in the exhaust is less than 1 per cent, it is very unlikely that there is any hydrogen or hydrocarbon, because CO is the least readily burned of the fuel gas constituents.

The following relation will be used to get most of the formulas.

Weight or volume of the substance sought, per cubic foot of fuel gas = weight or volume of that substance contained in 1 cu. ft. of exhaust gas  $\times$  volume of exhaust gas per cubic foot of fuel gas.

If an expression for the last named quantity, which is  $V_a$ , be deduced, it will remain only to find the required quantities per cubic foot of exhaust gas.

The following additional notation will be used.

$c, h, g$  = the number of molecule-feet in 100 molecule-feet of the fuel gas, of carbon, hydrogen, and oxygen, respectively; counted as in columns 3, 4 and 5 of the table on page 257.

$R$  = Ratio, by volume, of air supplied to gas, that is, the number of cubic feet of air per cu. ft. fuel.

To find  $V_a$ . Since there are  $D+M$  molecule-feet of carbon in 100 of the exhaust gas, and  $c$  molecule-feet in 100 of the fuel gas,

Cubic feet of exhaust gas per cubic foot of gaseous C in it =  $\frac{100}{D+M}$

Cubic feet of fuel gas per cubic foot of gaseous C in it =  $\frac{100}{c}$

By assumption, the gaseous carbon is the same in both gases. Hence, by division,

Cubic feet of exhaust gas per cubic foot of fuel gas =

$$V_a = \frac{c}{D+M}$$

The volumes of exhaust gas above are for dry gas.

The volume of air supplied per cubic foot of fuel gas is found as follows. The oxygen appearing in 100 molecule-feet of the exhaust gas is  $D+O+.5M$  molecule-feet. Dividing this by .21, the proportion of oxygen in air by volume, gives the corresponding amount of air; and the cubic feet of air which supplied this oxygen, per cubic foot of exhaust gas, is  $\frac{D+O+.5M}{100 \times .21}$ .

Multiplying by  $V_a$ , gives the cubic feet of air per cubic foot of fuel, that came in with the oxygen that *appears* in the exhaust gas. But some oxygen has disappeared in the form of water. As there are  $h$  molecule-feet of hydrogen in 100 of the fuel, and as each one combines with half its volume of  $O_2$ ,  $\frac{h}{2 \times 100}$  is the cubic feet of oxygen disappearing as water, per cubic foot of fuel. But some of the oxygen comes from the fuel itself, and we wish only that which comes from the air. Subtracting the molecule-feet of oxygen in the fuel gives  $(.5h - g) \div 100$ . Dividing by

O.21, we get the corresponding air. Then the cubic feet of air supplied per cubic foot of fuel gas is,

$$R = \frac{V_d}{21}(D + O + .5M) + \frac{.5h - g}{21}$$

If there is  $H$  per cent of hydrogen in the exhaust, too much oxygen has been counted. To allow for this, if the free hydrogen is known, subtract  $0.5H$  from the quantity in the parenthesis.

To find  $W_d$ . The density of  $\text{CO}_2$  in pounds per cubic foot is  $0.00559 \times \frac{44}{2}$  (see page 132). Multiplying this by  $\frac{D}{100}$  gives the weight of the  $\text{CO}_2$  in one cubic foot of the exhaust gas. In the same way the weights of all the other constituents of the dry exhaust gas may be obtained, and there results

$$\frac{.00559}{2}(44D + 32O + 28M + 28N) \div 100.$$

Multiplying this by  $V_d$ , and simplifying, we have,

$$W_d = \frac{V_d}{9000}\{11D + 8O + 7(M + N)\}.$$

To find  $W_w$ . Besides the water of combustion, there is some water vapor in the exhaust due to moisture in the air and fuel gas. These will be disregarded as they have but a slight effect upon the heat analysis owing to the fact that latent heat is not lost to the  $\text{H}_2\text{O}$  of humidity.

Since there are  $\frac{h}{100}$  cu. ft. of hydrogen per cubic foot of fuel, and since the weight of the resulting  $\text{H}_2\text{O}$  is 9 times the weight of the  $\text{H}_2$ , we have,

$$\begin{aligned} W_w &= 9 \times \frac{h \times .00559}{100} \\ &= .0005h. \end{aligned}$$

If hydrogen in the exhaust has been analyzed,

$$W_s = .0005(h - HV_d)$$

since free hydrogen would mean a corresponding decrease of  $H_2O$

The cubic feet of unburned CO per cubic foot of fuel equal

$$V_t = MV_d \div 100.$$

If hydrogen has been found, it may be expressed similarly

To find the excess coefficient, the value of  $R$ , as deduced above, should be divided by the expression for the air required as given on page 171, or

$$X = \frac{21R}{c + .5h - g}$$

**Problem 38<sub>1</sub>.** Draw a set of curves like Fig. 71 for the semi-anthracite coal whose ultimate analysis is given on page 147.

**Problem 38<sub>2</sub>.** How many pounds of air are required to burn 1 lb. of the coal in the last problem? In this calculation, what per cent of error is involved by ignoring the oxygen contained in the coal? *Ans.*, 10.95 lbs., 0.5%.

**Problem 38<sub>3</sub>.** How many cubic feet of air are required to burn 1 cu. ft. of the producer gas analysis of which is given on page 149? What is the weight of this air? *Ans.*, 1.07 cu. ft.

**Problem 38<sub>4</sub>.** Figure the density of the producer gas of the last problem by the molecular weight method, page 177. *Ans.*, .07 lbs.

**Problem 38<sub>5</sub>.** Deduce an expression for  $W_d$  similar to  $W_a$ , for coal.

**Problem 38<sub>6</sub>.** An exhaust gas analysis (coal) gives  $CO_2$ , 6 per cent;  $O_2$ , 13 per cent; CO, 1 per cent;  $N_2$ , 80 per cent. What are the weight and volume of free oxygen in it per pound of carbon gasified? Why would not a calculation for the air supplied, based on this result, give the same value as one based on the  $N_2$  in the exhaust gas? *Ans.*, 4.95 lbs.; 55.4 cu. ft.

**Problem 38<sub>7</sub>.** An exhaust gas analysis from the illuminating gas in the table on page 149, gives  $CO_2$ , 5 per cent,  $O_2$ , 10 per cent; CO, 1 per cent;  $N_2$ , 84 per cent. What are the values of  $R$ ,  $W_d$ ,  $W_s$ ,  $V_t$ , and  $X$ ?

## PART THREE

### THE TESTING OF POWER PLANT UNITS

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#### 39. THE DETERMINATION OF CYLINDER CLEARANCE

**Principles.** Linear clearance of a piston engine may be defined as the least distance between the piston and cylinder head in a direction parallel to the center line of the cylinder, when the engine is on dead center. It may have different values at the head end and crank end of the stroke.

Volumetric clearance is the cylinder volume between the piston and nearer cylinder end, which may be occupied by the working medium when the engine is on dead center. It may have different values on the two ends. The volumetric clearance is not only that in the bore of the cylinder, but includes the volume of the ports up to the valve face and the volume of any fittings, such as indicator piping, which may be filled with the working medium in the usual operation of the engine. The space in cylinder drain pipes open to the cylinder is sometimes considered as clearance volume; in some cases, however, this space, in usual operation, is filled with water of condensation so that it is not part of the clearance.

Volumetric clearance is expressed as a part or per cent of the piston displacement.

“Piston displacement” is the volume swept through by

one stroke of the piston, that is, the area of the piston times the length of stroke.

A knowledge of the clearance of engines is necessary to an analysis of their performance and losses, especially in connection with indicator diagrams. See Tests 42 (a) and (b).

**(a) Linear and Volumetric Clearance by Linear Measurements.**

The most accurate way to determine linear clearance, if the engine is not too large, is to loosen the connecting rod when the piston is at the end of its stroke, and then note how far the piston rod moves when it is pushed from the dead center position to contact with the cylinder cover. If the engine is too large for this the cylinder cover may be removed, and the measurement made by compressing between it and the piston, a small quantity of putty. The putty should be stuck to the cover at the part where the distance between it and the piston, or the piston rod extension, appears to be least. The piston should first be oiled and then dusted with graphite, so that the putty will not stick to it. The putty is then compressed by bolting the cover in place. The least thickness of the putty is the linear clearance. It may be measured by the use of a piece of fine, stiff wire.

The volumetric clearance may be obtained by mensuration from the drawings of the engine. This gives only an approximate result since the actual castings may differ materially from their drawings, and since the position of the piston in the cylinder may change with wear of the connecting rod bearings. If mensuration is used, it is best to make drawings of the engine clearances from the engine as it stands, as nearly as possible to scale.

**(b) Volumetric Clearance by Water Measurement.** The engine should be put on the desired dead center by putting the crank and connecting rod in line by eye. The steam valve or valves should be made tight either by smearing its face with heavy cylinder oil or, with slide valves, by squeezing a piece

of rubber gasket between the valve face and seat. The piston and cylinder bore should be well oiled to prevent leakage. If the cylinder has holes tapped on the top for indicator piping, they may be used for the introduction of water by which the clearance volume may be completely filled. By taking the weight of the water before and after filling, the amount necessary to fill the clearance space is determined. Dividing this by the density of the water gives the clearance volume. The density may be taken as 0.0361 lb. per cubic inch at 60° F. If the temperature is materially different from this, it should be noted and the corresponding density used.

Should there be a slight amount of leakage which cannot be prevented, it may be corrected for by determining the "leakage rate." This is done by measuring the amount of water necessary to keep the clearance space full for a period of one minute. Now, this is probably the maximum rate of leakage, since, during the filling, the leakage probably varied with the head of water in the cylinder. The average leakage may be taken as one-half that shown by the test. The period of time necessary for the filling should therefore be multiplied by one-half the maximum leakage rate to get the total leakage that occurred during filling. This weight subtracted from the weight used in the filling gives the weight corresponding to the actual clearance volume.

It should be noted that there are several assumptions in the foregoing which make the method approximate; it should not be used when the leakage is very large.

If the cylinder to be tested contains pockets in which air may be trapped during the filling, as is likely to be the case with internal combustion engines, these pockets should be blanked off and measured separately if possible. If this is not possible, the water method should not be used.

For accurate results, several determinations should be made and none accepted unless they show fair agreement.

To express the clearance volume in percentage, the piston displacement must be known. The bore may be measured with inside calipers, and the stroke by measuring the distance between dead center positions of the cross-head, a mark being made on the cross-head guides at each end to indicate these positions.

(c) **Volumetric Clearance from the Indicator Diagram.** Professor Paul Clayton proposed a method based on the fact that gases and vapors expanding or being compressed in closed cylinders follow the law that

$$\text{Absolute pressure} \times \text{volume}^n = \text{a constant.}$$

This law is represented by a logarithmic curve. Consequently if the amounts of the pressures and volumes obeying it are plotted on logarithmic coordinating paper (the coordinates being proportional to the logarithms of their markings as with the scale of a slide-rule), the result is a straight line. In an indicator diagram, the volumes represented are the total volumes of the working medium behind the piston, which include that of the clearance space. In order to transfer the indicator diagram to logarithmic coordinates, then, it is necessary to know the clearance volume so that the axis of abscissas may be located. If the indicator diagram is located on logarithmic coordinates by assuming a value of the clearance volume, the resulting expansion and compression curves will be concave to the origin, if the assumed value is too small, and convex if too large. This provides a method of determining the clearance volume, since it is only necessary to plot the expansion or compression curve from the indicator diagram on the logarithmic scale using a number of assumed values of the clearance volume until that one is found which produces a straight line. A convenient procedure is as follows.

Divide the indicator diagram with equidistant vertical lines into ten spaces, and mark the points of intersection of the verticals



with the expansion curve 1, 2, 3, etc. The coordinates of these points will be used to locate the logarithmic diagram. The absolute pressures corresponding to these points should be measured from the zero pressure line which is laid off 14.7 lbs. below the atmosphere line. After choosing a suitable scale with which to represent these pressures on the logarithmic chart, their positions are indicated by placing small marks against the vertical axis of logarithmic coordinates. Now, let a value of the clearance volume,  $x$ , be assumed in per cent. Then the point, 1, corresponds to a volume of 10 plus  $x$  per cent; point 2, to a volume of 20 plus  $x$  per cent, and so on. Using these volumes expressed as percentages, the points are readily located on the logarithmic chart. If the resulting curve is concave to the origin, a larger value of the clearance is assumed, and so on until that value which produces a straight line is found.

It will be noted that there are limiting values of the assumed clearance between which the straight line condition seems to be satisfied. On this account the method is recommended only where the clearance is large, as in internal combustion engines; the percentage of error then being not excessive.

**Problem 39<sub>1</sub>.** Following are the data from a test for volumetric clearance by the water method. Weight of bucket and water before filling, 12 lbs. 14 $\frac{1}{2}$  ozs. Weight after filling, 10 lbs. 2 $\frac{1}{2}$  ozs. Time of filling, 95 seconds. To keep clearance space full for 60 seconds, 8 ozs. What is the average leakage rate in pounds per minute? What is the clearance in cubic inches allowing for leakage? What is the per cent of clearance if the engine is 10 ins.  $\times$  16 ins. (bore  $\times$  stroke)? *Ans., 4.1%.*

**Problem 39<sub>2</sub>.** In the preceding problem room temperature 60° F. of the water is assumed. How much percentage of error would be involved if the actual temperature were 40° F.? If 80°? *Ans., 0.1%, 0.3%.*

## 40. VALVE SETTING OF A SIMPLE SLIDE VALVE ENGINE

**Principles.** It is assumed that the student is acquainted with the principles and operation of the simple slide valve and linkage and with the various quantities pertaining, such as steam lap, exhaust lap, angular advance, etc., which subjects are amply covered by various works on the steam engine.

With a finished steam engine, in order to set the slide valve so as to better the distribution of steam, there are only two parts that may be adjusted without altering the design of the parts, namely, the valve stem, which may be changed in length, and the eccentric, which may be changed in its position on the shaft relative to the crank. The former adjustment affects the laps and lead of the valve; the latter, the angular advance. A study of the valve motion will show the effects of such adjustments upon the valve and upon the steam distribution as follows.

On	Effect of Increasing.			
	Angular Advance.		Stem Length.	
	Head End.	Crank End.	Head End.	Crank End.
Outside lap.....	Same	Same	Increases	Decreases
Inside lap.....	Same	Same	Decreases	Increases
Lead.....	Increases	Increases	Decreases	Increases
Admission.....	Earlier	Earlier	Later	Earlier
Cut-off.....	Earlier	Earlier	Earlier	Later
Release.....	Earlier	Earlier	Earlier	Later
Compression.....	Earlier	Earlier	Later	Earlier

The student should confirm this table by drawing the linkage as in Fig. 77 to represent the positions at the various events of the stroke, taking them first for one end of the cylinder and then the other, and observing from the drawings the effects of

the adjustments as tabulated. In Fig. 77, the engine is shown at crank end release. The valve stem is broken so as to show the valve on top of the cylinder, for convenience, instead of at the side.

It should be noted that the laps may be measured when the eccentric is vertical, as this position of the eccentric puts the valve practically in its mid-position.

(a) **Measurement of Lead.** This is made by measuring the amount of port opening, the valve chest cover being removed, when the engine is on dead center, that is, when the crank and connecting rod are in line. The engine should not be put on dead center by noting the extreme position of the cross-head

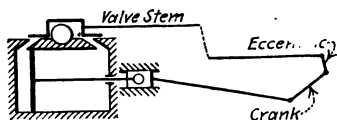


FIG. 77.

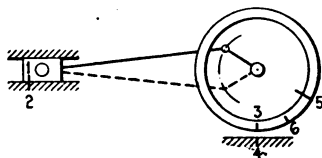


FIG. 78.

because, at the end of the stroke, there is no appreciable movement of the cross-head when the crank moves through several degrees. The eccentric, however, is then in a position where a few degrees of error will cause a considerable error in the position of the valve. Consequently, the dead center must be accurately located. The following method may be used. (See Fig. 78.)

The crank is turned until it is about 30 degrees from the dead center position sought. A mark, 1, is then made on the cross-head against a mark, 2, on the guides. Another mark, 3, is made on the flywheel against a mark, 4, on some stationary object. Now the engine is turned through its dead center position until the cross-head mark 1 again comes into the coincidence with 2. The crank and connecting rod will then occupy the positions shown by the dotted lines and the mark 3 will be at

5. A new mark is now made on the flywheel opposite 4, and the distance between this and 5 on the flywheel rim is bisected by the line 6. If the flywheel is then turned until 6 is opposite 4, the engine will be on dead center. Care should be taken that the motion of the linkage, when advancing to the positions for marking, should always be in the direction of the dead center sought so as to avoid error through lost motion of the parts. If there is no place conveniently near on which to put the mark 4, a pair of trammels should be used.

(b) **Setting the Valve for Equal Leads.** From the table given under "principles," it is seen a change in the angular advance will make the leads on both ends larger or both smaller, while a change of the stem length will make the lead on one end larger and on the other smaller. It will therefore be convenient to change the stem length to *equalize* the leads, and then to adjust the angular advance until they become the desired *amount*. The procedure is as follows.

The eccentric is first put in its approximate position, that is, about 135 degrees ahead of the crank. The leads are then measured according to (a). If the lead on the head end is greater than on the crank end, the stem length must be increased to equalize them, and vice versa (see table). If the leads are then both too small, the eccentric is shifted on the shaft further away from the crank until the desired leads are obtained, and vice versa if the leads are too large.

The proper amount of lead depends upon the travel of the valve, the engine speed, and the amount of compression. More lead is required for high speed engines, and less when the compression is high. For low speed engines, leads up to  $\frac{1}{16}$  in. are usual, and for high speeds, twice that amount.

It is advisable, after setting for equal leads, to indicate the engine as a check upon the setting. The lead should be such as to give a vertical line at the admission for both head and crank ends.

*Crank Position for Admission - 01*  
*" " " Cut-Off - 02*  
*" " " Release - 03*  
*" " " Compression - 04*

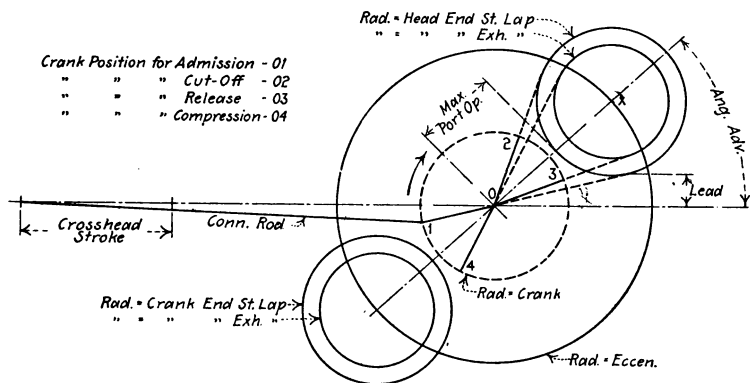


FIG. 79.—Bilgram Valve Diagram.

The values of the steam laps will depend upon the setting of the valve, but we may locate the lap circles by determining the sum of the steam laps. This equals the length of the valve

in the direction of its motion minus the distance between the outside edges of the ports, in the case of the D-valve taking steam outside. The steam lap circles may be constructed by first locating a point  $x$  on the eccentric circle such that the sum of its perpendicular distances from the lines 01 and 02 equals the sum of the steam laps. The lap circles may now be drawn about  $x$  as a center and tangent to the lines 01 and 02. The leads and maximum port openings are then determined as shown. If the leads are too large, a later cut-off must be assumed and the construction repeated.

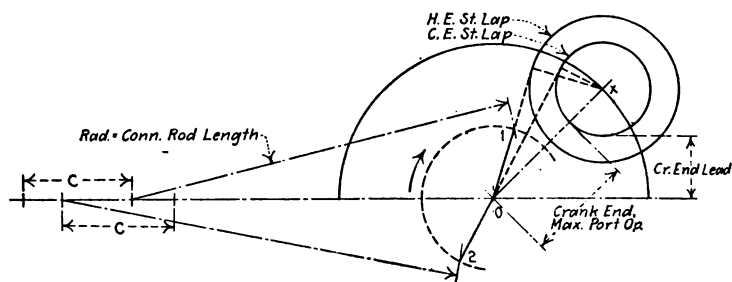


FIG. 80.—Setting for Equal Cut-offs.

Using the data for the lead and port opening at one end (preferably the crank end, since there they are larger) the valve may now be set for the assumed cut-offs. To do this, the crank is revolved until the valve has moved as far as it will toward the head end of the cylinder, thus giving the maximum port opening on the crank end and making the eccentricity parallel to the engine center line. The stem length is then adjusted until the crank end port is open an amount equal to that indicated by the diagram. Next the engine is put on its crank end dead center by the method given under (a) and the lead made equal to that shown by the diagram by swinging the eccentric on the shaft. This completes the setting, but the lead and port opening

should be measured on the head end as a check, and compared with the values given by the diagram.

(d) **Valve Setting by the Indicator.** The ultimate test by which the steam distribution must be judged is by the indicator. The valve setting should be such that not only does the engine run smoothly, but that it should give the maximum power for the steam used.

Valves may be set by deductions from the indicator diagrams, and the previously described methods dispensed with. The procedure may be more laborious and less systematic, since it is largely cut and try.

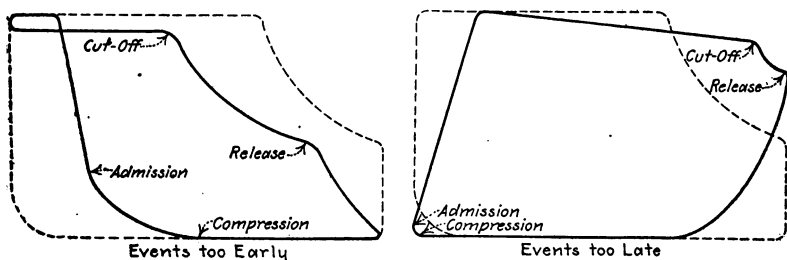


FIG. 81.—Faulty Valve Setting shown by Indicator Diagrams.

Fig. 81 shows indicator diagrams, superposed on ideal ones represented by dotted lines, from which may be noted the variations caused by too early and too late occurrence of the events of the stroke.

If the admission is too early (that is, if there is too much lead), full boiler pressure is admitted to the cylinder before the beginning of the stroke and the admission line rises at a point other than the extreme of the diagram. If the admission is too late (too little lead), the admission line slants inward because the full boiler pressure does not reach the cylinder until the piston has advanced appreciably in its forward stroke.

The cut-off may be too early on either end of the cylinder only in relation to that on the other end; they should be approximately equal for equal division of the load. The simple slide valve engine does not cut off earlier than six-tenths of the stroke; if the cut-off on either end is much later than this, as shown by the point at which the expansion line begins on the diagram, either the valve is not well designed or its setting is imperfect.

If release takes place too early, the pressure is released from the cylinder before the end of the stroke; the result is a curtailing of the expansion line and a sharp drop down to the backpressure line. If the release is too late, expansion may take place clear to the end of the stroke, but when the piston returns there is not enough opening for the steam to exhaust through freely. The result is excessive back pressure at the beginning of the return stroke as shown by the rounded toe of the diagram in Fig. 81.

The compressions on the two ends should begin at approximately the same percentage of the stroke. If there is too little compression, the engine will not run smoothly; if too much, the maximum power is reduced, although the economy in the use of steam may be increased. Compression tends to reduce the steam loss due to clearance.

To set a slide valve by the indicator, it is necessary to have diagrams from both ends of the cylinder. Faulty steam distribution may then be determined in detail by the effects on the diagrams as just described. A knowledge of the effects of the adjustments of stem length and eccentric position, as given in the table on page 190, then enables

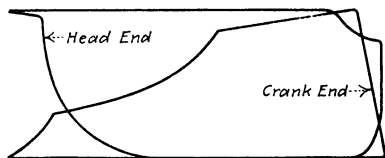


FIG. 82.

one to correct the steam distribution. The engine should then be indicated again, and further corrections made if necessary.



As an example of the reasoning involved, consider Fig. 82. It is well first to make a table showing the existing characteristics of the diagrams, as follows:

	Admission.	Cut-off.	Release.	Compression.
Head end.....	Early	Late	Late	Early
Crank end.....	Late	Early	Early	Late

By reference to the table on page 190, it is seen that to change the angular advance will aggravate some of the errors, whichever way it is adjusted. But if the stem length is increased, the events will tend to equalize and change to proper amounts. The stem length should therefore be increased and diagrams taken, the procedure being repeated until the cut-offs are about equal. It may then develop that all of the events are too early or too late, in which case the eccentric should be adjusted until the admission lines are vertical. If the release and compression are then satisfactory, the setting is acceptable.

(e) **Study of the Steam Distribution by the Bilgram or Zeuner Diagrams.** This may be done before or after setting the valve in order to decide upon the best setting or how the setting may be improved, thus possibly saving a number of trial settings and of indicator trials. For this purpose, the necessary measurements are made of the valve and engine parts, from which a number of diagrams are constructed with different adjustments of eccentric and stem length assumed. Indicator diagrams may be constructed from the data yielded by the Bilgram or Zeuner diagrams (the expansion and compression curves being drawn as rectangular hyperbolas), by comparison of which, the best setting may be selected.

The measurements to be made from the engine are of the connecting rod and crank lengths, the eccentricity, and the lead and outside and inside lap on one end. The outside lap at either end may be readily determined by noting the distance the valve must move from mid-position to begin to uncover the port.

The mid-position may be found by making a prick point or scratch on the valve stem and measuring to this mark. The inside lap at either end may be obtained by subtracting from the length of the valve face, the length of the port and the outside lap, the dimensions being taken from the end considered. The sum of the outside laps should also be found as described under (c), and the sum of the inside laps, found similarly, by subtracting the distance between the inside edges of the valve faces from the distance between the inside edges of the ports.

The Bilgram diagram may now be laid out by drawing first the eccentric circle and the lead at the end for which it has been measured, say the head. (See Fig. 79.) Next, the head end lap circle is drawn with its center at such a point,  $x$ , on the eccentric circle as to make the lap circle tangent to the lead line. The angular advance line is then drawn in, which locates the centers

of the other lap circles. Having described these the events of the stroke, in percentages, may be determined.

The Zeuner diagram may be laid out similarly, starting with the eccentric circle (see Fig. 83). A point  $X$  is then located at a distance from the center of the eccentric circle equal to the lap plus the lead on the head end. The circle  $OXA$  is next constructed.

which fixes the angular advance by the line  $OA$ . When the lap circles are put in, the events of the stroke may be determined in the usual manner.

It may be noted that it is convenient to work with the sum

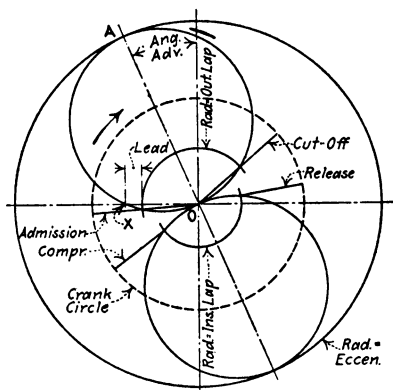


FIG. 83.—Zeuner Valve Diagram.

of the outside laps and of the inside laps, since, if it is desired to reconstruct the diagram with a different lap on one end, the opposite one is readily determined by subtraction.

**Problem 40<sub>1</sub>.** Verify the table on page 190 by the method given in the text following it.

**Problem 40<sub>2</sub>.** The lead on the head end of a steam engine is found to be  $\frac{1}{16}$  in., and on the crank end  $\frac{3}{16}$  in. If the leads are to be made equal to  $\frac{1}{16}$  in., should the valve stem be lengthened or shortened and how much? Should the angular advance be increased or decreased?

**Problem 40<sub>3</sub>.** A D-slide valve has a face 1.5 ins. long at each end, and the distance between the faces is 2.75 ins. The ports are each  $\frac{1}{4}$  in., and the outside edges of the ports are 4.25 ins. apart. If the head end outside lap is .625 in., what are the values of the other three laps?

*Ans.,  $\frac{5}{8}$ ,  $\frac{1}{8}$ ,  $\frac{3}{8}$  in.*

**Problem 40<sub>4</sub>.** The valve in the preceding problem has an eccentricity of 1.375 ins. Find the lead and the maximum port opening on the crank end for equal cut-offs, using the Bilgram diagram. The connecting rod is 28 ins. long and the crank 5 ins.

**Problem 40<sub>5</sub>.** Find the events of the stroke in per cents using the data of the last problem and the Bilgram diagram.

**Problem 40<sub>6</sub>.** Repeat the last problem using the Zeuner diagram.

**Problem 40<sub>7</sub>.** Why cannot a valve be set for equal leads and equal cut-offs at the same time? If for equal leads, which cut-off is greater?

**Problem 40<sub>8</sub>.** How could the events of the stroke be measured direct from an engine whose valve was set, by reference to the valve and measurements of the cross-head travel?

**Problem 40<sub>9</sub>.** Set the valve of an engine for equal cut-offs. Then measure the events of the stroke from the engine, from the indicator diagram actually obtained, and from the Bilgram diagram.

#### 41. THE MECHANICAL EFFICIENCY TEST OF A STEAM ENGINE.

**Principles.** The mechanical efficiency of a steam engine is equal to the useful horse-power divided by the horse-power developed by the steam in the cylinder. If a prony brake is used to measure the useful horse-power, then (see Test 6)

$$\text{B.h.p.} = BWN$$

is the useful horsepower.

Using the following notation,

I.h.p. = Indicated horse-power,

$P_h$  and  $P_c$  = the mean effective pressure on the head end and crank end respectively,

$L$  = the length of the stroke in feet,

$a$  = the area of the cylinder, in square inches,

$a'$  = the area of piston rod, in square inches,

$N$  = the number of revolutions per minute,

then the cylinder horse-power may be expressed,

$$\text{On the head end, I.h.p.}_h = \frac{P_h L a N}{33,000} = K_h P_h N. \quad (1)$$

$$\text{On the crank end, I.h.p.}_c = \frac{P_c L (a - a') N}{33,000} = K_c P_c N. \quad (2)$$

$K_h$  and  $K_c$  are called the "engine constants."

$$\text{Total I.h.p.} = \text{I.h.p.}_h + \text{I.h.p.}_c.$$

Mechanical friction causes a loss of power so that all the work developed in the cylinder does not appear at the flywheel. The friction horse-power equals

$$\text{F.h.p.} = \text{I.h.p.} - \text{B.h.p.} \quad (3)$$

When the brake is entirely removed from the engine, the work done in the cylinder to keep the engine running is expended to overcome friction only.

The function of the engine governor is to keep the speed constant. This is done by the action of any change of speed, through centrifugal force, to alter the position of the governor weight. More or less steam is admitted into the cylinder according to the position of this weight, and therefore more or less cylinder work is done. To accommodate itself to a variable

load, then, the engine *must* allow some change of speed to regulate the cylinder work. At no load, with the governor admitting the least steam, the speed is greatest, and at maximum load, least. Good regulation requires that the difference between these limits of speed be small.

For comparative purposes the speed regulation is expressed as a percentage variation from the mean speed. If  $N_m$  and  $N_n$  are the speeds at maximum and no load, respectively, then

$$\text{Per cent speed regulation} = \frac{(N_n - N_m) \div 2}{(N_n + N_m) \div 2} \times 100.$$

(a) **The Indicated Horse-power** is determined by measuring the quantities in formulas 1 and 2, the engine constants being calculated before the test. The mean effective pressures are found from indicator diagrams by integration with a planimeter. The speed is determined with a hand or continuous counter.

The brake horse-power being the independent variable, it is first calculated what net forces should be obtained at the brake scales to produce the required horse-powers. The engine is then operated at one of these horse-powers by adjusting the tension of the brake strap to give the necessary force at the scales, and to maintain it at that value. This means constant observation of the brake scales and occasional regulation of the brake strap tension. The engine should be operated at each load for ten minutes or more so that it may adjust itself to the new conditions of load and friction. A number of indicator diagrams and measurements of the R.p.m. are then taken. It is convenient to average the readings of  $P_h$ ,  $P_e$ , and  $N$  at each load and to use the averages for substitution in formulas 1 and 2.

A more convenient, although approximate, formula for calculating the indicated horse-power, is as follows.

$$\text{I.h.p.} = k(P_h + P_e)N$$

in which  $k$  is the average of the two engine constants or  $L\left(a - \frac{a'}{2}\right) \div 33,000$ . A brief consideration will show that the more nearly is the area on the crank end equal to that on the head (that is, the smaller is the area of the piston rod relative to that of the cylinder) the less error is there in the approximate formula. Also, the less difference is there between the mean effective pressures on the two sides of the piston, the less error is there. When they are equal the results from the two methods of calculation are the same.

The following is a rational formula by which the per cent of error ensuing from the approximate formula may be determined for any conditions. Let

$R$  = Ratio of  $P_c$  to  $P_h$ .

$d$  = Diameter of the piston rod, inches.

$D$  = Diameter of the cylinder bore, inches.

Then

$$\text{Per cent of error} = 50 \times \frac{d^2}{D^2} \times \frac{1-R}{1+R}.$$

Using this formula, it will be found with usual proportions of  $d$  and  $D$ , that the mean effective pressure on one end must be two or three times that on the other to produce more than one per cent of error. It is thus clear that the approximate calculation may almost always be used, since any engine with properly set valve would give a more uniform division of the load than this under normal conditions. Some engines divide the cylinder work quite unequally at light loads; in such a case the formula will show whether or not the correct method should be used. A convenient rule is as follows:

*If the mean effective pressure on one end is not greater than*

$$\frac{50d^2 + D^2}{50d^2 - D^2},$$

*times the mean effective pressure on the other, then the error is less than one per cent.*

(b) **The Friction Horse-power** at each load is found as indicated by formula 3. Tests have shown that this quantity is very nearly constant at all loads of a given engine, if properly operated. As the conditions of lubrication affect friction and therefore mechanical efficiency, the lubrication should be given special attention and maintained uniform throughout the test. It is advisable to plot a curve of F.h.p. vs. B.h.p. as the test progresses as this is closely indicative of the accuracy of the results.

(c) **The Efficiency** is found by dividing each value of the B.h.p. by the corresponding value of the I.h.p.

(d) **Speed Regulation.** Values of  $N_m$  and  $N_n$  have been obtained for the other results, and these may be used to get the speed regulation. Another test consists in quickly throwing the entire load on or off and noting the resulting momentary variation in speed. This is greater than that produced by a gradual change of the load, and it should be determined with a chronograph or tachograph.

Any change in boiler pressure will affect the speed regulation, so this should be kept constant during the test if possible, and any variations noted.

**Problem 41<sub>1</sub>.** If the F.h.p. is constant at all values of the B.h.p., deduce the forms of the curves of B.h.p. vs. I.h.p. and efficiency vs. B.h.p. Will these curves pass through the origin or not? Why?

**Problem 41<sub>2</sub>.** Figure what may be the maximum ratio of mean-effective pressures with a 14-in.  $\times$  30-in. engine having a  $2\frac{1}{8}$ -inch piston rod, so that the error from using the approximate formula for I.h.p. will be less than 1 per cent.

*Ans., 5, nearly.*

**Problem 41<sub>3</sub>.** Calculate the length of the adjustable arm of a polar planimeter, so that horse-power may be read direct when indicating the engine in the preceding problem. (See Test 14 (e).) The diameter of the record wheel is 0.79 in. and there are 100 graduations on the wheel. One graduation to equal 1 H.p. Length of indicator diagram = 5 in. Spring scale = 60. Average value of  $N$  is 80.

*Ans., 3.62 in.*

**Problem 41.** A 50-horse-power engine running at 200 R.p.m. is tested with a brake with an 8-ft. arm. If its unbalanced weight is 10 lbs. and the weight of the pedestal by which its thrust is transmitted to a platform scales is 5 lbs., what should be the scale readings for an efficiency test of six runs?

*Ans.*, 179 lbs., max.

**Problem 41.** Run a series of tests on a steam engine to show the relation between steam pressure at admission if the governor is throttling, or cut-offs, if of the cut-off type, under variable brake horse-power.

## 42. ECONOMY TEST OF A STEAM ENGINE

**Principles.** Economy tests on steam engines are made to determine the amount of steam and heat they consume per unit of power, under different conditions. The most important variable in such tests is the brake horse-power; it is usual to vary it throughout the working range for a complete economy test.

Economy results should always be based on the brake horse-power, but the indicated horse-power is often used because it is inconvenient or impossible to brake the engine. The unit "horse-power-hour" will be used, meaning the amount of work developed in one hour by one horse-power. The heat equivalent of this work should be remembered. Since 33,000 foot-pounds of work are developed in one minute by one horse-power and 778 foot-pounds equal one B.t.u., then in one hour

$$\frac{33,000}{778} \times 60 = 2545 \text{ B.t.u.} = 1 \text{ H.p.-hr.}$$

The steam supplied to the engine is generally expressed in pounds per horse-power-hour, that is, the pounds of steam supplied in one hour divided by the horse-power. The result should be considered in connection with the condition of the steam, since less will be needed at high pressure or superheat, and more if it is wet. One practice is to quote the steam consumption in pounds of "equivalent dry steam," that is, the pounds



of dry steam that contain the same number of heat units as the total  $H_2O$  actually supplied. Since these amounts of steam have not by any means equivalent capacities for doing work, and since the costs of making wet, saturated, and superheated steam of the same heat content are not the same, this seems hardly worth while. It is recommended, instead, to include a statement of the condition of the steam in the expression for steam consumption.

The heat consumed by the engine may be figured from the total weight of steam supplied and the heat *consumed* from each pound. This latter expression requires consideration. Each pound of steam received by the engine carries heat in the form of latent heat and heat of the liquid. Part of this total heat is converted into useful work and part is lost to friction, radiation, etc., in the cylinder; the balance is rejected. During passage through the cylinder, part of each pound of  $H_2O$  originally supplied is condensed, so that when it appears in the exhaust, it is partly water and partly steam at a considerably reduced pressure. The heat content of such a mixture is  $h' + x'l'$  (see page 125 for notation), so that, properly speaking, the heat consumed by the engine is  $H - (h' + x'l')$ , in which  $H$  is the total heat of the steam supplied. That is, the heat consumed is the difference between the heat supplied and the heat rejected. Of the heat rejected, however, the latent heat cannot be made use of without bringing in auxiliary apparatus such as a feed-water heater. Therefore it is reasonable to charge the latent heat of the exhaust against the engine. On the other hand, the heat of the liquid of the exhaust may be reclaimed if the exhaust is condensed and returned to the boiler. Therefore, a fair, though arbitrary, standard is

$$H - h'$$

for the heat consumed per pound of steam. Under ideal conditions the exhaust could be returned to the boiler as feed water

without any drop of temperature;  $h'$  may therefore be taken as the heat of the liquid corresponding to the pressure or temperature of the exhaust. This is the British standard and is to be preferred. The steam engine code of the A.S.M.E. recommends that  $h$  be the heat of the feed water actually returned but as this depends upon the power plant and not the steam engine, it seems less logical to use where results for the engine only are sought.

If  $S$  is the weight of steam supplied per horse-power-hour in pounds, then the heat consumed per horse-power-hour is

$$S(H - h').$$

(a) **Steam Consumption.** Having measured the horse-power, it is only necessary to get the hourly rate of steam supplied. This may be done in various ways as follows.

**By Indicator Diagram.** The engine itself makes a crude form of steam meter since at every stroke a definite volume of steam is taken into the cylinder which can be measured from the indicator diagram. Referring to Fig. 30, page 47, at the point  $b$  cut-off takes place, the cylinder is closed to the boiler, and the cycle commences with a volume of steam equal to that represented by the point  $b$ . This comprises the clearance volume and the volume of the piston displacement up to the point  $b$ , or  $(c + C) D$  cubic feet;  $c$  being the clearance expressed as a part of the piston displacement;  $C$ , the cut-off expressed as part of the stroke, and  $D$  the piston displacement in cubic feet. Assuming the steam having this volume to be saturated, we may find its density in pounds per cubic foot from its pressure. Calling the density  $W$ , we have  $W(c + C)D$  lbs. as the weight of the steam at the point  $b$ . Not all of this steam has been furnished by the boiler at the beginning of the cycle, however, since some steam from the previous stroke was compressed in the clearance space. Just before the valve opened to the boiler,

The steam in the cylinder had a pressure and volume corresponding to the point *e*, Fig. 30. Its weight then is  $wcD$ . The amount of steam furnished by the boiler to one end each revolution is therefore  $W(c+C)D - wcD$  lbs. If  $N$  is the revolutions per minute, the weight per hour is

$$60 ND\{W(c+C) - wc\}.$$

This is the weight used on one end of the cylinder on the assumption that the steam is saturated just after cut-off. The weight on the other end may be found similarly. Now, the steam is *not* dry because of initial condensation, and it may have varying degrees of wetness depending upon the cut-off, the working range of temperatures, type of engine, etc. Numerous empirical formulas have been proposed for the calculation of cylinder condensation, an important item, since for simple engines it amounts to between 20 and 50 per cent of the amount of steam shown by the diagram. A method has been proposed by Professor J. Paul Clayton which takes advantage of the law discovered by him that there is a definite relation between the amount of cylinder condensation and the exponent of expansion in the equation  $PV^n = \text{a constant}$ , at all events for certain types of engines. (See Journal A.S.M.E., April, 1912.) He plots the indicator diagram on logarithmic coordinates (see Test 39 (c)) and from the slant of the expansion line gets the value of  $n$ . By using a set of curves giving the quality of the steam in the cylinder just after cut-off at various values of pressure and  $n$ , the quality of the steam at cut-off and the cylinder condensation may be calculated. These curves have not been sufficiently verified as yet to make them of general application. Furthermore, the method seems to take no account of leakage of steam past the valve, which may never enter the cylinder, but is nevertheless consumed by the engine.

The clearance should be determined as described under

**Test 39.** The expression for cut-off,  $C$ , may be found by dividing the distance of  $b$ , Fig. 30, from  $ae$ , by the length of the diagram, in inches.

For the sampling of diagrams, see Test 10 (e).

**By Condenser.** All of the steam used is passed into a surface condenser, and the condensate weighed for a counted time. This is probably the most accurate way of testing for steam consumption. Numerous time-quantity readings should be made to determine uniformity of flow. The condenser must be operated with more circulating water than is used in practice, so that the condensate will emerge cool; otherwise a large amount may escape unweighed by evaporation. The piping between the condenser and engine must be examined for tightness, and the condenser tested for leakage. The latter may be done by running the air-pump when no steam is flowing, and noting if any circulating water is drawn through, preferably when the condenser is hot. If it leaks a small amount, a leakage rate may be found and applied as a correction to the results. If the engine is to be tested "non-condensing," that is, with no vacuum, the condenser must be vented in order to establish this condition.

**By Steam Meter.** One of the commercial forms of steam meter may be used in the steam pipe supplying the engine. See Test 28.

**By Feed Water Measurement.** For this method, the engine and boiler supplying it should be isolated so that all of the steam generated by the boiler is used in the engine. Sometimes it is necessary to supply steam to auxiliary apparatus, such as a feed pump, from the boiler supplying the engine tested. In such a case, it is necessary to make a separate measurement of this steam, either by establishing the rate beforehand or, better, by condensing the steam used by the auxiliary during the test. It is necessary to examine the boiler and piping for tightness, the latter especially at branches stopped by valves.

This is done by closing all valves in branches and the main stop valve at the engine so that the supply pipe is open from the boiler to the engine valve, but closed everywhere else. With a quiet furnace fire so that there is no active evaporation, the level is then noted in the water column at a number of uniform intervals of time. If the level falls, leakage is taking place, and the rate should be determined from the area of water surface calculated from the measurements of the boiler. This leakage may then be applied to the results of a test as a correction.

The feed water may be measured by any of the methods given under Test 50 (b).

**Willan's Law** states that the weight of steam per unit of time used by an engine with a throttling governor varies directly as the indicated horse-power of the engine, very nearly. The work represented by an indicator diagram in which the expansion follows the law that  $PV = a$  constant (very nearly the case with steam) is mathematically proportional to the initial pressure, cut-off being constant. As the pressure is proportional to the density of the steam, approximately, it follows that the weight of the steam is proportional to the work, and the indicated horse-power. The weight of steam used also varies directly with the brake horse-power, since  $B.h.p. = I.h.p.$ —a constant. Experiment has shown that this relation applies not only to throttling engines but to those of the cut-off type, and to steam turbines.

The practical application of Willan's law lies in the consequence that if the weight of steam used per hour by an engine is plotted against its indicated or brake horse-power, the result is a straight line. This furnishes a check upon the results of a test as the test proceeds; if points so plotted do not follow a straight line, there is error. It should be noted that at overloads, the curve is apt to deviate slightly from straightness, and lean toward the axis of steam weights.

**The duration of the test** should depend upon the uniformity of conditions, the capacity of the engine, and the method used

for measurement of the steam. During each test at a given load, weighings are made at uniform time intervals, say ten or fifteen minutes. When there are four to six of these of nearly equal amounts recorded consecutively, the run may be discontinued, provided the error of starting and stopping is within reasonable limits. (See page 9.) This error depends upon the method of measuring the steam. If a condenser is used, the error will be relatively small, since it equals the difference in the amounts of condensate in the condenser at starting and stopping. If the boiler method is used, the test must be much longer as there may be large error in the level of the water column, due to differences of density, ebullition, etc. With a steam meter, the test need be only long enough to get sufficient indicator diagrams for a fair average, provided the engine has been previously given a settling run.

Determinations of the condition of the steam supplied as to quality and pressure, and of the pressure or temperature of the exhaust, should be made and included in the statement of steam consumption per horse-power-hour.

The relation between the steam consumption in pounds per indicated horse-power-hour,  $S_1$ , and per brake horse-power-hour,  $S_2$ , is

$$S_1 = S_2 \times \text{mechanical efficiency.}$$

(b) **Cylinder Condensation** is often figured by subtracting the total steam used per hour, as shown by the indicator diagram, from that determined by direct measurement. The difference includes not only cylinder condensation, but valve leakage in the case of engines which do not have separate valves controlling the exhaust.

(c) **Thermal Efficiency.** There are two standards, one having a commercial, the other a scientific value. The former is the ratio of the heat equivalent of the useful work to the heat units consumed as defined under "principles." Since there are

$S(H-h')$  heat units consumed for every horse-power-hour of useful work, and since the heat equivalent of a horse-power-hour is 2545 B.t.u.,

$$E = \text{thermal efficiency} = \frac{2545}{S(H-h')},$$

in which  $S$  may be based on indicated or brake horse-power. If the former, the result is the "cylinder efficiency"; if the latter, it is the "over-all efficiency."

$H$  may be obtained from the pressure and quality determinations of the steam supplied, by use of the steam tables;  $h$  is determined similarly from readings of a pressure gage or thermometer in the exhaust.

The scientific standard is the Clausius efficiency, which is the ratio of the heat converted into work by the actual engine to that by an ideal one working without clearance space and with non-heat-conducting cylinder walls with cut-off early enough to allow expansion clear down to the back pressure. There will then be no clearance loss and the expansion will be adiabatic. The indicator diagram representing these conditions is shown by Fig. 84, and is that of the Clausius cycle although it is more frequently referred to as the Rankine cycle. The initial pressure and quality of the steam and the back pressure are assumed to be the same in the ideal engine as in the actual. This is a more reasonable standard for efficiency than the commercial standard since the latter charges against the engine heat that it could not use under even ideal circumstances because of the limitations in the operation of the working medium.

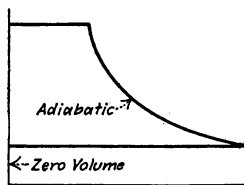


FIG. 84.—Clausius Cycle.

To figure the Clausius efficiency it is first necessary to get

the heat units that are converted into useful work from 1 lb. of steam under actual conditions. This is

$$E(H-h').$$

Under the conditions of the assumed ideal engine the heat converted into work is the difference between the total heat  $H$  of the steam supplied and the total heat of the exhaust  $H_1$ , since there are no losses.  $H_1$  is the theoretical amount of heat left in a pound of steam after it has expanded adiabatically from the given pressure and condition as to quality to the given back pressure. It may be calculated by the use of entropy tables, but it is more convenient to get it directly from the Mollier total heat-entropy diagram which gives the heat of steam under all conditions and at various stages of adiabatic expansion. Having found  $H_1$ , the Clausius efficiency is obtained from the relation

$$E_c = \frac{E(H-h')}{H-H_1}.$$

Another way of figuring this efficiency is to divide the steam consumption of the ideal engine,  $S_c$ , in pounds per horse-power-hour, by that of the actual engine,  $S$ .

$$E_c = \frac{S_c}{S} = \frac{2545}{H-H_1} \times \frac{1}{S}.$$

This is a useful form since by it, the cylinder or over-all efficiency may be readily obtained, depending upon the basis of  $S$ .

**Problem 42<sub>1</sub>.** An indicator diagram shows the cut-off to be one-fifth of the stroke. The pressure at cut-off is 110 lbs. abs.; at the end of compression it is 25 lbs. abs. The engine is 10 in.  $\times$  12 in.  $\times$  100 R.p.m., single acting with 4 per cent clearance. How many pounds of steam are supplied per hour, allowing 25 per cent of that shown by the diagram for condensation?



**Problem 42.** If the mean-effective pressure in the preceding is 25 lbs., what is the steam consumption in pounds per I.h.p.-hour? *Ans.*, 39.2.

**Problem 42.** If the steam in the supply pipe (last problem) is at 105 lbs. gage and contains 3 per cent of moisture, and if the exhaust is at 3 lbs. gage, what is the thermal efficiency? *Ans.*, 6.69%.

**Problem 42.** What is the Clausius efficiency for the foregoing? *Ans.*, 46.4%.

**Problem 42.** Deduce from the typical form of Willan's line, the form of the curve between steam consumption in pounds per B.h.p.-hour and B.h.p. What is the value of the steam consumption when B.h.p.=0?

### 43. ECONOMY TEST OF A STEAM TURBINE

**Principles.** The measurements and results, in general, are the same as those for a reciprocating engine, except that there can be no indicated horse-power determination, and often no brake measurements, an electrical load being considered instead. Since it is not possible to indicate a turbine, the term "internal horse-power" is sometimes used to express the equivalent of indicated horse-power, but it is not a definite quantity. It is often impracticable to apply a brake to the turbine shaft on account of the direct connection of a generator; in such a case the turbine and generator must be tested as a single unit. For other details, see Test 42, principles.

(a) **Determination of the Useful Horse-power** may be made the same as for Test 41 if the turbine alone is tested with the use of a brake. Otherwise, the electrical load should be measured in kilowatts by taking either wattmeter readings or readings of amperes and volts. The horse-power equivalent to the electrical output, that is, the "electrical horse-power," may be found from

$$1 \text{ Electrical horse-power (E.h.p.)} = 0.746 \text{ kilowatts.}$$

For alternating current generators, the code of the A.S.M.E. states that the electrical output is the kilowatts delivered to the switch-board less that required for field excitation when the field is

separately excited. Under this condition, therefore, measurements of the field current and voltage should be made.

(b) **Steam Consumption.** The pounds of steam consumed per hour may be measured by any of the methods described under Test 42 (a) except by indicator diagram. Dividing this by the kilowatts or horse-power gives the pounds of steam per kilowatt- or horse-power-hour.

It is often desirable, for purposes of comparison, to base the steam consumption on brake horse-power or "internal horse-power." The former quantity may be estimated, if an efficiency curve of the generator is available, by multiplying the steam consumption in pounds per electrical horse-power-hour at a given load by the generator efficiency at that load.

The internal horse-power is difficult to estimate with any degree of accuracy. It is sometimes assumed to be the quotient of the brake horse-power and the mechanical efficiency of a steam engine working under the same conditions. This, of course, is a crude estimate. Using it, however, the steam consumption may be expressed on the basis of internal horse-power.

For duration of the test, see page 209.

**Additional Data.** If there are traps arranged to catch condensation from the turbine casing, they should be drained regularly and the condensate weighed in with the steam consumed. Readings should be made of the pressure in the nozzle chamber to show the drop of pressure through the governor valve. If this drop is excessive, the steam consumption will be correspondingly high. The predetermined conditions of pressure, quality, and back pressure or vacuum should be kept as constant as possible, and the quantities carefully measured, as they have a decided influence upon the economy of a turbine. It is important that the barometer be read to get the required accuracy in the measurement of vacuum which should be expressed as an absolute pressure. This is because at low pressures, the heat content of steam varies materially with the pressure, and also

because turbine economy is more dependent upon the predetermined vacuum than pressure or superheat.

For the purpose of estimating the steam consumption of a turbine under a given set of conditions of steam pressure, quality and vacuum, such as are stated in the manufacturer's guarantee, the turbine being tested under somewhat different conditions, correction curves are supplied by the manufacturer. These show the number of pounds of steam per horse-power- or kilowatt-hour to be subtracted from the test result if the pressure is higher than that stated, the number to be subtracted if the superheat is higher, and the number to be subtracted if the back pressure is higher, or vice versa.

Willan's line should be plotted during all trials under variable output.

(c) **The Thermal Efficiencies** are obtained exactly as for Test 42 (c), the basis for the computation of useful work being the electrical, brake, or internal horse-power.

(d) **Separation of Losses.** The losses of a turbine may be considered in two groups. First, approximately constant losses which include those due to friction and radiation. Frictional resistance is encountered at the bearings and stuffing boxes, and between the discs and blades and the steam, and through windage of the external parts. Second, steam losses including leakage through clearance spaces, condensation, and faulty action of the blades and nozzles in not completely absorbing the energy of the steam.

These losses may not be separately measured without an involved test. Professor Carpenter has indicated a method of separating the two groups by the use of Willan's line. When plotted against brake horse-powers, this line intercepts the axis of steam consumption at a point which shows the steam consumed per hour to overcome the first group of losses, that is, at no load. The distance parallel to the load axis, measured in horse-power, necessary to move Willan's line so that it shall pass

through the origin, equals the power given to the first group of losses. At any other load, this quantity is to be subtracted from the power that would be developed by the actual amount of steam consumed if operating on the Clausius cycle; the result is the actual brake horse-power plus the second group of losses, from which the second group may be readily obtained.

**Problem 43<sub>1</sub>.** A turbine, tested at full load, consumes 3150 lbs. of steam in 90 minutes. The current delivered is 340 amperes and 220 volts. The generator efficiency at full load is 95 per cent and the friction losses of the turbine are 5 per cent of the power delivered to the generator. What is the steam consumption in pounds per kilowatt-hour, per E.h.p.-hour, per B.h.p.-hour, and per internal horse-power-hour?

*Ans.*, 28.1 per Kw.-hr.

**Problem 43<sub>2</sub>.** For the test of Problem 43<sub>1</sub>, average values as follows were obtained. Steam pressure, 150 lbs. gage; superheat, 80° F.; vacuum, 20 inches mercury; barometer, 28.5 inches. What would have been the steam consumption if the following conditions held? Steam pressure, 180 lbs., abs.; superheat, 100° F.; vacuum, 28 ins.; barometer, 30 ins. Corrections are 0.05 lb. per kilowatt-hour for each pound difference in the steam pressure, 0.02 lb. for each degree of superheat, and 1.0 lb. for each inch of vacuum.

*Ans.*, 20.5 lbs. per Kw.-hr.

**Problem 43<sub>3</sub>.** What is the efficiency (British standard) for the results of Problem 43<sub>1</sub>? How much efficiency should be added or subtracted for each unit of steam pressure, back pressure, and superheat. (See Problem 43<sub>2</sub>.)

#### 44. ECONOMY TEST OF A STEAM POWER PLANT

**Principles.** A complete test consists of measurements of the amounts of heat distributed and lost in the entire system, beginning with the energy of the fuel and ending with the energy delivered at the shaft or switchboard. For the present we shall consider only that part of the plant beyond the boiler, the subject of boiler trials being separately treated under Test 50.

The most important result from power plant tests is the fuel consumption in pounds of fuel per horse-power- or kilowatt-hour. Other results are the steam and heat consumption of the engine and the auxiliaries, and the heat balance. The latter is an equation between the total heat available and the various

amounts accounted for in its distribution. They may be found in heat units per hour and expressed as per cents of the total heat per hour added to the feed water entering the boiler. If these percentages are separately multiplied by the boiler efficiency, the items of the heat balance will then be percentages of the heat available in the fuel. This follows from the fact that boiler efficiency is that part of the fuel energy which is added to the feed water and distributed as steam.

The boiler trial, which is part of the complete test and conducted at the same time, gives the efficiency of the boiler and its losses, all based on the heat value of the fuel. A complete heat balance is thereby obtainable.

The usual condensing power plant contains as auxiliaries a condenser, circulating pump, air pump, and feed water pump. There may also be a cooling tower and its auxiliaries, and a feed water heater using the exhaust steam from the auxiliaries, and such minor apparatus as separators, traps, etc., which may or may not be arranged to drain back to the boiler.

The rearrangement of the apparatus and piping for the purpose of testing should interfere as little as possible with normal operating conditions, and any departures from them that may effect the results, such as different feed water temperatures, should be allowed for by making a separate test to measure the actual working quantities that have been altered.

Equipments of different power plants are too varied for very specific directions for the measurements of the various quantities. Any of the methods given under Test 42 (a) for the measurement of steam weights should be applied according to the exigencies of the particular test. Any one item of the heat balance may be omitted from the direct measurements, since one item may always be figured by subtraction.

The basis of all heat measurements is the heat contained by the feed water just before it enters the boiler or economizer if there is one.

**Duration.** If a boiler trial is included, its duration determines the length of the test; if the engine and auxiliaries only are to be tested, the duration should be the same as for a test of the engine alone. See Tests 42 and 50, principles.

**(a) Steam and Heat Consumption of Engine and Auxiliaries per Horse-power or Kilowatt-hour.** Let  $w$  stand for weight of steam per hour;  $H$ , for its total heat per pound above 32 degrees depending upon its pressure and quality; and let  $h$  be the number of heat units per pound of the feed water above 32 degrees. Let the subscripts  $e$ ,  $c$ ,  $a$ ,  $f$ , and  $b$  refer to the engine, circulating pump, air pump, feed pump, and boiler respectively. Then the steam consumption is

$$(w_e + w_c + w_a + w_f) \div \text{horse-power or kilowatts output.}$$

To get the heat consumption, the various values of  $H$  should be used for strict accuracy, as determined by the pressure and quality of the steam just before it is delivered to each unit, but in most cases the following form is practically correct.

$$\text{Heat consumed per hour} = w_e(H_e - h) + (w_c + w_a + w_f)(H_b - h).$$

Dividing this by the horse-power or kilowatts gives the required quantity.

The heat consumed per hour by the engine and auxiliaries may also be determined by subtracting the heat lost to leakage and drips from the total heat added to the feed water (see (b) and (c) following).

The values of  $H$  should be obtained from the steam tables by reference to the measurements of pressure, temperature, and quality of the steam at the point considered. For  $H_e$  this should be immediately beyond the main stop valve of the boiler; for  $H_b$ , it should be just behind the main stop valve of the engine, between it and the separator if one is used.  $h$  is obtained from the temperature of the feed water just before it enters the boiler. If an injector is used, the temperature should be taken before

entering the injector, since the heat added by that instrument comes from the boiler and returns to it. The heat consumed by the feed pump in this case should be figured separately and is very nearly equal to the heat equivalent of the work done in pumping. (See Test 48.)

**(b) Heat Added per Hour to the Feed Water.** If there is only one source of supply yielding  $W$  lbs. per hour, then the heat added is

$$W(H_b - h).$$

If in addition to this there is another supply such as would be obtained from traps, separators, etc., draining back to the boiler in an independent pipe carrying  $W_2$  lbs. per hour, then the heat added is

$$W_1(H_b - h_1) + W_2(H_b - h_2),$$

the subscripts 1 and 2 referring to the main and auxiliary supplies, respectively.

The quantities  $H_b$  and  $h$  are measured as under (a). The weight of the main supply is generally measured by a weighing system as for boiler trials (see Test 50 (b)). Auxiliary supplies of feed may be metered or directly weighed during a special test made for that purpose.

**(c) Heat Lost to Leakage and Drains.** Leakage may occur from the steam space of the boiler or from the steam pipes either to the air or to pipe branches, and it may occur below the water level in the boiler. Leakage which cannot be prevented should be measured by a preliminary test similar to that described on page 209 for leakage correction. It is assumed that the rate of leakage thus determined remains the same through the test of the plant. Calling it  $w_l$  pounds per hour, the heat lost is

$$w_l(H_b - h)$$

If the separators, traps, etc., do not drain back to the boiler, they may be kept closed during the test and drained at regular intervals into buckets of cold water whereby they are weighed. If  $w_a$  is the number of pounds of water withdrawn per hour, the loss is

$$w_a(H_b - h)$$

If this water is returned as an auxiliary supply,  $h_2$  should be substituted for  $h$  in the above.

(d) **Heat Balance.** All the items, having been measured as above in heat units per hour, may now be expressed as percentages of the heat added to the feed water. Then the heat balance is

$$\begin{aligned} 100\% &= \% \text{ of heat consumed by engine} \\ &+ \% \text{ of heat consumed by auxiliaries (stated separately or collectively)} \\ &+ \% \text{ of heat lost to leakage and drains.} \end{aligned}$$

The first item on the right-hand side may be subdivided into the per cent of heat equivalent to the useful work and the per cents representing the various engine losses.

It should be noted that this method of analyzing the heat distribution charges against the engine and auxiliaries the losses of radiation between the condenser and the boiler. Also, if a feed water heater is used, the heat consumed by the engine figures less, although the actual performance of the engine is exactly the same as if no heater were used, because the value of  $h$ , the heat of the feed water, is higher. The only way in which the effect of the heater appears numerically is in the heat added to the feed water, but no idea of the saving due to the heater may be formed.

**Problem 44.** A condensing plant is arranged so that the exhaust from the auxiliaries preheats the feed water after it leaves the condenser, in a closed heater. Tell how the plant could be tested so as to show the loss due to drop in the heat of the liquid between the engine exhaust and the heater entrance, and the gain due to the feed water heater.



## 45. VALVE SETTING OF A DUPLEX (STEAM) PUMP

**Principles.** The duplex pump consists of two reciprocating steam pumps operating side by side and arranged so that the piston rod motion of one operates the steam valve of the other. Each pump consists of a water cylinder whose piston is directly driven by the piston rod of the steam cylinder.

The steam valves distribute the steam so that one pump is on its forward stroke when the other is on its return. The valves have neither laps nor lead, so that cut-off occurs simultaneously with release. The valve stem of each pump is driven by a rocker deriving its motion from the piston rod of the other pump (see Fig. 85). As each rocker moves through the whole piston rod stroke and as it is desirable that the valves throw only at the end of the stroke, a certain amount of lost motion is provided between the valve stem and the valve.

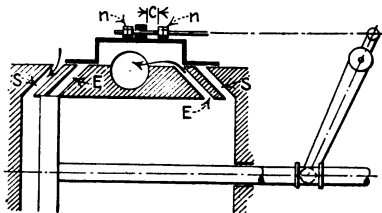


FIG. 85.—Duplex Pump Valve Motion.

This is shown by the distance  $c$  in the figure. The valve stem passes freely through a hole in a lug on the valve, and gives motion to the valve only when the nuts  $n$  and  $n$  come into contact with the lug. By adjusting the lost motion, the valve may have more or less throw.

There are two steam and two exhaust ports for each steam cylinder,  $SS$  and  $EE$ . This is arranged as a safeguard against the piston hitting the cylinder head. If the valve should not close the exhaust port before the end of the exhaust stroke, the piston itself will close it thus entraining steam in the clearance space which acts as a cushion.

(a) **Adjustment of Lost Motion of Valve.** This is accomplished by setting the valves in their mid-positions when the pistons are in their mid-positions.

The pistons should be set by reference to prick marks on the piston rods. The limiting positions of stroke should be measured by prying each piston rod with a bar to the extreme of its travel, pressure behind the pistons being released by opening drains or otherwise. When the piston rods are set midway between these positions, the rockers should be vertical.

With the steam chest cover removed, the valves are now placed so that they completely cover the ports. The nuts *n*, *n*, Fig. 85, are then adjusted so that there is an equal amount of clearance between each one and the valve lug. With some designs, there is only one nut to be adjusted on each stem, this nut then being between two lugs. In such a case the clearances may not be changed in amount, but only equalized. In other designs there is a "lost motion link" outside of the valve chest by which the clearances may be adjusted without opening the chest.

The proper amount of clearance depends upon the size and design of the pump and may be determined by trial. If the steam ports are to be opened full each stroke, then the total lost motion should be equal to the throw of the valve stem corresponding to normal piston stroke less twice the steam port length parallel to the stroke.

(b) **Readjustment after Trial.** If, after the valves are set, the pump runs "lame," that is, the strokes are unequal in length or time, it may be because the resistances of the water cylinders are different. They should then be examined for leaky water valves, tight stuffing boxes, etc. Such faults being corrected, if the pump still runs lame, the lost motion of the valves must be readjusted until the desired equality of strokes is obtained. It need only be remembered that to diminish the lost motion makes the valve throw greater, thus providing better access to the steam and release to the exhaust.

**Problem 45<sub>1</sub>.** Explain why the ordinary duplex pump cannot use steam expansively.

**Problem 45<sub>2</sub>.** If the stroke of one piston rod of a duplex pump were curtailed at the head end, being normal in other particulars, what should be done to the valve motion to remedy it?

#### 46. MECHANICAL EFFICIENCY TEST OF A RECIPROCATING STEAM PUMP

**Principles.** The mechanical efficiency of a steam pump equals the useful or water horse-power, W.h.p., divided by the indicated horse-power of the steam cylinders. If  $D$  is the total head in feet pumped against, including suction and discharge, and  $W$  is the number of pounds of water delivered per minute, then,

$$\text{W.h.p.} = \frac{WD}{33,000}.$$

If there were no leakage of water in the pump from one side of the piston to the other or through the valves, then the work done per minute would be equal to the product of the total pressure pumped against in pounds per square foot and the volume displaced by the piston in cubic feet per minute. If  $p$  stands for the pressure in the discharge pipe;  $p'$ , for the vacuum in the suction pipe, both in pounds per square inch; and if  $L$  and  $A$  are the same as in the formula for indicated horse-power, and  $N$  is the number of working strokes per minute, then

$$\text{W.h.p.} = \frac{144(p+p')LAN}{33,000 \times 144}.$$

Leakage of water in a pump, of the kind referred to, is called "slip." It is generally expressed numerically as a percentage of the piston displacement.

(a) **Steam End Indicated Horse-power at Various Water Horse-powers.** The indicated horse-power is found exactly as for a steam engine (see Test 41 (a)), except in regard to measurement of the stroke length,  $L$ . With direct acting reciprocating pumps, there is no constant limit to the stroke length and in ordinary operation it may vary materially. An average value should therefore be obtained from a number of measurements throughout the test. There is on the market a continuous recording device for this purpose, but if one is not available, the average stroke may be obtained from the average lengths of the indicator diagrams, the ratio of reduction being known. Another method is to set a scale against a mark on some projecting part of the piston rod, and to note the travel of this mark at regular time intervals.

The water horse-power may be varied either by changing the speed of the pump or the discharge pressure. With reciprocating steam pumps, the pressure is usually approximately constant, and the water load varies with the quantity of water demanded. Either the pressure or the speed may be the independent variable during test, according to the operating conditions of the pump.

The pressure may be varied during the test for different runs, by opening the stop valve in the discharge pipe more or less.

The quantity of water delivered per minute may be measured by any of the methods of determining water rates. For large pumps, weirs may be used.

The head against which the water is discharged may be measured with a pressure gage set in the discharge pipe. The suction head generally may be measured in feet, and taken as the difference in level between that of the water supplied and the center of the pressure gage used for measuring discharge pressure. Since 1 lb. per square inch = 2.3 feet head, the total head is

$$D = 2.3p + D'$$

in which  $D'$  is the suction head in feet.

Readings of  $p$  and  $D'$  should be taken at regular intervals during each run.

These quantities determine the water horse-power according to the formula previously given. The water horse-power may be estimated, according to the second formula, if the discharge pressure and the number of strokes per minute are measured, and if a value for the percentage of slip is assumed (see (d)).

Neither of the formulas for water horse-power takes into account the kinetic energy of the water delivered. Generally the velocities of the water are low enough to make this a negligible quantity. If it is desired to calculate the horse-power due to kinetic energy, the velocity should first be figured from the cubic feet discharge per unit of time and the cross-section of the pipe. The result is then obtained by,

$$\text{W.h.p. available from kinetic energy} = \frac{Wv^2}{2g} \div 33,000$$

in which  $W$  has the same value as before,  $v$  is the velocity in feet per second, and  $g = 32.2$ , nearly.

**(b) Mechanical and Fluid Losses.** The loss due to mechanical friction of the pump parts may be obtained by indicating both the water and steam ends of the pump. The indicated horse-power of the water end is obtained in exactly the same way as for the steam cylinder, the same values being used for the number of strokes per minute and the average length of stroke. Then the steam end I.h.p. minus the water end I.h.p. equals the mechanical friction horse-power.

The fluid losses are due to leakage, or slip, fluid friction of the water against its passage in the ports, eddies, etc. Slip causes loss of power through water being pumped against pressure from one part of the pump to another without being discharged into the delivery main. These losses are included in the power shown by the indicator diagram for the water end.

Consequently the horse-power lost equals the I.h.p. of the water end minus the W.h.p. as calculated under (a).

(c) **The Mechanical Efficiency** for each value of W.h.p. is obtained by dividing the W.h.p. by the corresponding I.h.p. of the steam end. It is sufficient, for each run, to use average values of heads, water rates, mean effective pressures, and stroke lengths and speeds with which to calculate a single value of efficiency.

The indicated horse-power of the water cylinders is sometimes measured. This divided by the I.h.p. of the steam cylinders is the efficiency covering mechanical losses only.

(d) **Slip.** The piston displacement in cubic feet is  $\frac{LAN}{144}$  per minute. The water actually displaced, in the same units, is  $W/62.5$  at ordinary temperatures. The percentage of slip is therefore,

$$\frac{\frac{LAN}{144} - \frac{W}{62.5}}{\frac{LAN}{144}} \times 100 = \left(1 - \frac{2.3W}{LAN}\right) \times 100.$$

The quantities involved have been obtained for the other determinations.

(e) **The Capacity** of pumps is generally expressed in gallons per twenty-four hours. Knowing the weight of water, or cubic feet, discharged per minute, the capacity is readily calculated, for which purpose the following closely approximate relations may be used.

1 gal. weighs 7.5 lbs. 1 cu.ft. contains 8.33 gal.

**Problem 46.** The following data are obtained from the test of a duplex pump. Discharge pressure, gage, 35 lbs.; suction lift, 4.5 ft.; water discharged per minute, 110 lbs.; length of stroke, 5 ins.; number of strokes per minute, both cylinders, 250; size of water cylinders, 2 ins.  $\times$  5 ins. What is the W.h.p., the percentage of slip, and the capacity of the pump?

*Ans.*, 0.283 H.p.; 0.226%.

**Problem 46.** In the preceding problem, if the internal diameter of the discharge pipe is 1.38 ins., what is the horse-power due to velocity?

**Problem 46.** If the pressure in the discharge pipe is 12 lbs., and if the water supply is 6 feet above the gage indicating the discharge pressure, what is the total head?

*Ans.*, 21.6 ft.

**Problem 46.** Run a series of tests on a steam pump to show the relation between slip and discharge pressure, and slip and speed.

## 47. ECONOMY TEST OF A STEAM PUMP

**Principles.** These, in general, are the same as given for the economy test of a steam engine, Test 42, the only difference being that the useful work is measured in terms of water horse-power as defined under Test 46.

In addition to the measurements listed under Test 42, another one is usually required, namely, of "duty." Duty is the number of foot-pounds of useful work performed by a pump per million B.t.u. consumed by the engine.

(a) **Steam and Heat Consumption.** The pounds of steam consumed per hour may be found by any of the methods under Test 42 (a). The water horse-power is determined as under Test 46 (a). From these two quantities the pounds of steam per W.h.p.-hour may be calculated.

The heat consumed per pound of steam is  $H - h'$ , using the same notation and reasoning as given under Test 42, principles. Pressure readings of the supply and exhaust steam, and quality determinations of the supply steam are required for these heat quantities.

(b) **Thermal Efficiency.** This is the same as for Test 42 (c) except that the British standard only need be used in most cases.

(c) **Duty.** The thermal efficiency (British standard), when expressed as a fraction, is the heat equivalent of the useful work done per B.t.u. consumed. The number of foot-pounds of useful work done by each B.t.u. consumed is therefore 778 times the

efficiency, 778 being the mechanical equivalent of heat. Since the duty is the number of foot-pounds per million B.t.u. consumed,

$$\text{Duty} = 778,000,000 \times \text{Efficiency.}$$

It should be noted that the efficiency is here expressed not as a per cent, but as a fraction of the heat consumed.

**Problem 47<sub>1</sub>.** The steam used for the test cited in Problem 46<sub>1</sub> was 88 lbs. per hour, and was at 85 lbs. gage pressure, quality 97 per cent. The pump exhaust steam was at 10 lbs. gage. What were the steam consumption in pounds per horse-power-hour, the heat consumption in B.t.u. per horse-power-hour, the thermal efficiency, and the duty?

*Ans., Eff. = 0.86%.*

**Problem 47<sub>2</sub>.** A pump discharges 4750 lbs. of water against a total head of 100 ft., in a certain time. During the same time it is supplied with 100 lbs. of steam and from each pound it consumes 1000 B.t.u. What is the duty of the pump?

*Ans., 4,750,000.*

#### 48. ECONOMY TEST OF AN INJECTOR

**Principles.** The injector is a pump in which the heat energy of steam is directly used. Works on steam engines and steam boilers describe the instrument in detail. The results from an injector test are the same as those from a steam pump test. In addition, the number of pounds of water pumped per pound of steam supplied is usually quoted. The methods of testing are somewhat different, inasmuch as a direct measurement of the weight of steam per hour is not necessary, this quantity being obtained from the heat balance.

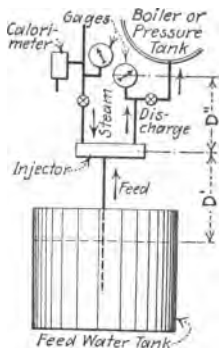


FIG. 86.—Injector Test.

Fig. 86 shows diagrammatically the arrangement of an injector for test.

An expression for the weight of steam used per minute is deduced by equating the heat energy of the



steam entering the injector, plus the mechanical and heat energy of the water entering, to the heat and mechanical energy of the water discharged. The datum of the heat measurements is at 32° F., and of the energy measurements is the level of the injector.

Let  $W$  = weights, in pounds per minute.  
 $h$  = heat of the liquid.  
 $L$  = heat of vaporization of the steam supplied.  
 $x$  = quality of the steam supplied, if wet.  
 $p$  = pressure of discharge, pounds per square inch.  
 $D', D''$  = suction head, and distance between injector and gage, feet, respectively.  
 $t$  = temperatures, degrees F.

Subscripts  $f, s, d$  refer to feed water, steam, and discharge, respectively.

Then, expressing all energies in heat units, and neglecting kinetic energy

$$W_f h_f - \frac{W_f D'}{778} + W_s (h_s + xL) = W_d h_d + \frac{W_d (2.3p + D'')}{778} + \text{Radiation.}$$

Also,

$$W_d = W_f + W_s.$$

Solving these equations simultaneously and neglecting radiation, we have

$$W_s = \frac{h_d - h_f + \frac{2.3p + D''}{778} + \frac{D'}{778}}{h_s + xL - h_d - \frac{2.3p + D''}{778}} \times W_f.$$

In this expression, the heat equivalents of the work are very small compared with the other quantities and they may be omitted. For the heat of the liquid,  $h$ , in each case may be substituted its closely approximate value,  $t - 32$ . There results,

$$W_s = \frac{t_s - t_f}{t_s + xL - t_d} \times W_f.$$

**Selection of the Independent Variable.** This may be the discharge pressure, the feed water temperature, the suction head, or the steam pressure. When the injector is used as an ordinary pump, the discharge pressure is perhaps the most important of these. If this is selected as the independent variable, it may be controlled by means of the valve in the discharge pipe, see Fig. 86. If the steam pressure is selected, it may be varied by the stop valve in the steam pipe, but the gage should be placed on the other side of the valve to record the pressure on the injector. Whichever quantity is selected, all of the others should be kept as constant as possible throughout each run, and the injector should be regulated before each run so as to secure maximum flow.

**(a) Pounds of Water Pumped per Pound of Steam.** It must first be decided whether to credit the injector with the feed water only, or with the feed water plus the condensed steam which appears in the discharge. If the injector is used for *emptying*, as when draining a mine, the former amount is logically used; if for *filling*, as when supplying a tank at a height, the latter amount is proper. When used as a boiler feed, there is room for some discussion of which is the correct amount. The condensed steam is returned to the boiler, but it is used again to operate the injector, so strictly speaking it is not a part of the useful feed water. On the other hand, a duplex pump used for feed water would *not* return the steam used to drive it and would have to pump that much more water.

From the equation previously deduced, we have

$$\frac{W_f}{W_s} = \frac{t_s + xL - t_a}{t_a - t_f},$$

which is the pounds of feed water pumped by one pound of steam. Adding one to the numerical value of this expression gives the pounds of water plus steam pumped.

The temperatures are obtained from thermometers appropriately placed in the discharge, feed, and steam pipe; the latent heat, from the steam tables; and the quality, from a calorimeter. It is often sufficient to assume a value for the quality, provided that may be done within 2 per cent of its true value. The ejector should be run long enough before each run at the required conditions to establish uniform temperatures. The run should be long enough to secure enough temperature readings for a fair average, and for the proper measurement of the quantities mentioned later.

(b) **Pounds of Steam per Water Horse-power-hour.** The equation for  $W_s$ , previously deduced, gives the pounds of steam used per minute. The water horse-power is

$$\text{W.h.p.} = \frac{W_f(2.3p + D' + D'')}{33,000} \quad \text{or} \quad \frac{W_f D' + W_s(2.3p + D'')}{33,000},$$

the one or the other to be used according to the considerations given under (a). The pounds of steam per water horse-power hour is then

$$S = \frac{60W_s}{\text{W.h.p.}}$$

For this quantity it is necessary to make the same measurements as enumerated under (a) and also to measure the feed water rate and the heads,  $D'$  and  $D''$ . The pounds of feed water per minute may be determined by any of the methods for measuring water. A convenient arrangement is to use a hook gage or water glass in the tank from which the feed is obtained, by which the drop in water level may be timed. This causes a variation in the suction head, but if the tank is of generous cross-section, the percentage of variation may be small enough not to effect the results materially. The average value of the suction head may be figured from the readings of water level. The head  $D''$  is, of course, constant throughout the test.

(c) **Thermal Efficiency.** When considered merely as a pump, the same expression for efficiency as for a steam engine holds, Test 42 (c).

$$\text{Efficiency} = \frac{2545}{S(h_s + xL - h_a)}$$

When considered as a boiler feed apparatus, the efficiency is much higher than this because the heat in the discharge is then "useful," and the only loss is that due to radiation. A determination of radiation may be made from the heat balance if the steam supplied is measured as well as the feed water, from which the boiler feed efficiency may be found. This requires much more precision in all the test measurements than the other results do, however, because the radiation is a small difference between two large quantities.

(d) **Duty.** See Test 47 (c).

(e) **Capacity** should be figured in gallons per twenty-four hours from the test results of  $W_f$  or  $W_a$ , according to the use to which the injector is to be put.

**Problem 48<sub>1</sub>.** A test of an injector yields the following data. Steam pressure 100 lbs. gage; quality, 97 per cent; temperature of feed, 60° F.; temperature of discharge, 147° F.; pressure of discharge 105 lbs.; gage in discharge pipe, 5 ft. above injector; level of feed water, 10 ft. above injector; cross-section of supply tank, 7.1 sq. ft.; rate of fall of water in supply tank while pumping, 4.12 in. per minute. What is the ratio of water pumped to steam used, the pounds of steam per W.h.p.-hour, the efficiency, and the duty? Base all results on the total amount of discharge. *Ans., Eff. = 0.378%.*

**Problem 48<sub>2</sub>.** What should be the capacity of an injector to supply a 100-H.p. boiler, if it operates only two-thirds of the time?

## 49. ECONOMY TEST OF THE PULSOMETER

**Principles.** The pulsometer is a pump in which the pressure energy of steam is applied directly to the water pumped without intermediate pistons. It consists of two cast iron chambers controlled by automatic valves and working alternately. Steam,

entering one, discharges the water by its superior pressure, while in the other, the partial vacuum, formed from steam condensed by contact with the water, draws a fresh supply of feed.

The results from a pulsometer test, the equations to be used, and the method of testing are exactly the same as for an injector test, 48, to which the reader is referred.

## 50. ECONOMY TEST OF A STEAM BOILER

**Principles.** The heat available in the coal consumed in the furnace of a steam boiler is distributed under four heads: First, the useful heat which goes to evaporate the feed water; second, the heat lost with the carbon slipping through the grate or removed in cleaning; third, the heat carried up the stack by the exhaust gases; and fourth, the heat lost by radiation and all otherwise unaccounted for losses. The heat carried up the stack may be traced under the subheads, heat carried as temperature of the dry exhaust gases, heat carried by steam formed by evaporation of water in the coal and from the combustion of hydrogen, and heat lost through incomplete combustion of carbon, hydrogen, and hydro-carbons.

A complete boiler test to determine these quantities includes measurements of feed water and coal supplied for a counted time, and analyses of the coal, ash, and flue gases.

Besides the heat balance, certain other quantities are to be found for the purpose of comparing the general performance of the boiler with that of others. The following terms should be understood in this connection.

A *unit of evaporation* (U.e.) is 970.4 B.t.u. added to the feed water. This unit equals the latent heat of steam at atmospheric pressure. A unit of evaporation will therefore make one pound of dry steam in a boiler operated at atmospheric pressure and

supplied with feed water at 212° F., or, more briefly, "from and at 212 degrees."

The *factor of evaporation* (F.e.) is the number of units of evaporation added to each pound of the feed water. If  $H$  is the total heat of 1 lb. of steam generated under the existing conditions of pressure and quality, and  $h'$  the heat of the feed water, both counted from 32 degrees, then the heat added per pound is  $H - h'$ , and

$$\text{Factor of evaporation} = \frac{H - h'}{970.4}.$$

*Equivalent evaporation* (E.e.) may be regarded as the number of units of evaporation added to a stated amount of feed water, and is expressed in units per hour, or per square foot of heating surface, or per pound of fuel, etc. Since each pound of feed water has F.e. units of evaporation added,

$$\text{E.e.} = \text{No. of pounds of feed water} \times \text{F.e.}$$

This quantity also equals the number of pounds of dry steam which would be generated by the transfer of the same quantity of heat in a boiler operating "from and at 212 degrees." Equivalent evaporation is generally referred to as a number of pounds of steam under these conditions.

A *boiler horse-power* (Bo.h.p.) is generated when 34.5 units of evaporation are transferred to the feed water per hour. Knowing the total number of units of evaporation developed during one hour in any boiler trial, the boiler horse-power is

$$\text{Bo.h.p.} = \frac{\text{U.e. per hour}}{34.5}$$

The numerator of this fraction is equivalent evaporation per hour. Therefore

$$\text{Bo.h.p.} = \frac{\text{Pounds of feed water per hour} \times \text{F.e.}}{34.5}$$

If in this expression we put  $\text{Bo.h.p.} = 1$ , then the pounds of feed water per hour equals  $\frac{34.5}{\text{F.e.}}$ . That is

*The number of pounds of feed water to be evaporated in one hour to make one boiler horse-power equals 34.5 divided by the factor of evaporation, the latter quantity being fixed by the conditions of feed water and steam.*

The rate of combustion for coal fired furnaces is the number of pounds of coal consumed per hour per square foot of grate.

**Duration of test** is governed by the error of starting and stopping. All the conditions of the boiler and furnace should be the same at the end of the test as they were at the beginning, especially in regard to the quantity and quality of coal on the grate, and water in the boiler. If the coal remaining on the grate at the end of the test is greater than that on the grate at the start, the furnace will be charged with a greater amount than it has actually consumed, and vice versa, unless an estimate is made of the difference between the amounts. The probable error of such an estimate is greater in the case of the coal than of the water. Consequently, the error in the coal measurement determines the duration of the test. The test should be long enough to make the error of starting and stopping less than 1 or 2 per cent of the total amount of coal fired. Assuming this error to correspond to a difference of 1 in. in the fuel bed, the resulting error in the coal quantity will be 2.5 lbs. for each square foot of grate surface, the weight of a cubic foot of incandescent coal being about 30 lbs. Two and one-half pounds is 1 per cent of 250 lbs., so the test should be continued until a total

of 250 lbs. of coal per square foot of grate surface has been fired. The higher the rate of combustion, the shorter is the duration. If the rate is 20, for example, the test should be about twelve hours long.

In all cases, the length of the test should be a multiple of the regular cleaning period in order to cover and keep constant regular operating conditions. In the example just cited, if the cleaning period is eight hours, the test should be continued for sixteen hours.

**Starting and Stopping.** The boiler should be operated during a preliminary run and complete readings taken until they show uniformity of all conditions. The furnace should then be thoroughly cleaned, enough live coal being left on the grate, as in usual cleaning, to start a new fire. The thickness of this coal bed should be estimated and noted, and the level of the water in the water column marked with a string tied around the glass. The boiler is then fired with a fresh charge of weighed coal, the time of firing the first shovelful of which is taken as the start of the main test. The ash-pit should then be cleaned and all temperatures, pressures, etc., read. At the end of the test, the cleaning is repeated in exactly the same way as at the beginning, the time of firing the first shovelful of the fresh coal after cleaning being taken as the end of the test. The same observations are made as at the beginning. It is best to note the water level just after the cleaning, with the fire door open, because then there will be a minimum of ebullition to cause a false level.

**Sampling** of the coal should be done as for the proximate analysis Test 35, page 149. Sampling of the flue gas is described under Test 38, page 173. Sampling of the ashes and refuse removed in cleaning and falling through the grate is according to the same principles as for coal. The steam should be sampled for quality determinations, page 129.

To make clear the quantities resulting from a boiler test, a set of calculations from observations made for an actual test



will be given for each quantity analyzed in the following. The condensed or average observations are given below.

# DATA FROM A BOILER TEST AND NOTATION USED IN FORMULAS

Duration of test, hours.....	= 12
Total weight of coal as fired, pounds.....	= 4925
Total weight ash and refuse, pounds.....	= 734
Total weight water evaporated, pounds.....	= 36,400
Boiler pressure, pounds per square inch, abs.....	= 110
Quality of steam, $x$ .....	= 0.992
$t_f$ = Temperature of feed water, degrees F.....	= 98
$t$ = temperature boiler room, degrees F.....	= 80
$T_g$ = temperature flue gas, degrees F.....	= 680
Proximate analysis. Percentages by weight of coal as fired.	
Moisture.....	= 2.75
Volatile matter.....	= 6.00
Fixed carbon.....	= 78.45
Ash.....	= 12.8
Weight of carbon in 1 lb. of refuse.....	= 0.20
$m, vm, fc$ = weights of moisture, volatile matter, and fixed carbon per pound of dry coal, respectively.	
Heat value of coal by calorimeter.....	= 13,040

## VOLUME ANALYSIS OF THE FLUE GAS, PER CENTS

$D$ = carbon dioxide.....	= 11.0
$O$ = oxygen.....	= 9.0
$M$ = carbon monoxide.....	= 0.5
$N$ = nitrogen.....	= 79.5

## WEIGHTS, IN POUNDS, OF CARBON FROM 1 LB. OF DRY COAL

$C_a$ = carbon wasted in ash;	
$C_t$ = total carbon; fixed, and in volatile matter;	
$C_g$ = carbon burned, appearing in flue gas.	

(a) **Determination of Total Coal and Refuse.** The coal to be fired is weighed out by the barrowful as it is needed and heaped on the boiler room floor at a place convenient for firing, one barrowful at a time. The following is the best form of log for

coal records. It includes a few observations to illustrate its purpose.

Weight of Barrow.		Net Weight.	Total Weight.	Time of Firing First Shovelful.
Empty.	Full.			
Test started..	.....	.....	.....	8:00 A.M.
100	250	150	150	8:00
100	275	175	325	8:24
100	260	160	485	8:46
Etc.				

A chart should be plotted as the test proceeds of total coal as shown by the fourth column against time as a base. It should be noted particularly that the time to be plotted against the total weight from each barrowful is not the time of firing the first shovelful from the corresponding weighing of coal, because this coal is not entirely burned until the first shovelful of the next charge is fired. Referring to the log, 150 lbs. should therefore be plotted against 8:24; 325 lbs. against 8:46; and so forth. If the rate of evaporation and all other conditions external to the furnace are uniform, then with proper firing, the rate of combustion will be uniform, and a fair straight line will be represented by the points of the plot. This line should pass through the origin; if it intersects the axis of coal weights above the time axis, it indicates that there was too much coal on the grates at the start; if below, too little. In such a case, it is better to figure the total coal from the slant of the line.

The ashes accumulating under the grates and the refuse removed during cleanings in the main test should be collected, allowed to cool without wetting, and then weighed.

(b) **Determination of Total Feed Water Evaporated.** The feed water may be measured by a meter in the feed water line, but when certainty of accurate results is desired, a direct weighing system such as represented diagrammatically by Fig. 87

should be used. With this arrangement, the levels of the water at the start of the test is marked by the gage, *g*, in the suction tank and by the string on the water column at *m*. The feed pump should be operated at a constant rate through the test, this rate being determined by the horse-power required of the boiler; and, with proper firing and regulation, the water level shown by the column should always be approximately at *m*. The best form of log for the records is as follows. A few sample observations are included to illustrate.

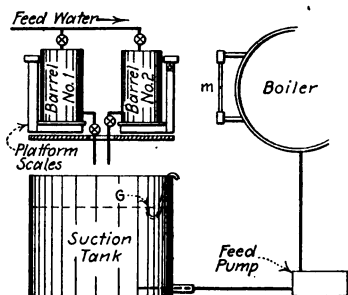


FIG. 87.—Weighing System.

Barrel No.	Weight of Barrel.		Net Weight.	Total Weight.	Time of Passing Gage Level.
	Empty. Lbs.	Full. Lbs.			
Test started.....					8:00 A.M.
1	50	300	250	250	8:12
2	60	320	260	510	8:25
1	75	300	225	735	8:34
Etc.					

As the test proceeds, a chart should be plotted of total water as shown by the fifth column against the time taken to evaporate it as a base. When the first charge, 250 lbs. on the log, is turned into the suction tank, the level is raised above the gage, *g*, Fig. 87. When the level has again descended to the gage, all of the 250 lbs. has then been evaporated provided that none of it has accumulated in the boiler as would be shown by a higher level in the column than *m*. The time of passing the gage level in the suction tank is therefore the time to be plotted; 250 lbs. against 8:12; 510 lbs. against 8:25; and so forth.

It is especially important to keep the rate of evaporation uniform for a valid test. The water-time curve should therefore be a straight line.

If the water level is not the same at the end as at the beginning of the test, the total water may be figured from the slant of the line, or a correction may be applied to the last figure in the fifth column of the log. The correction is calculated from the cubical contents of the boiler between the two levels and the density of the water at the existing temperature. It should be noted that a false level may appear in the water column if the ebullition is violent or if the water column is blown down within a short time before reading. The latter occurrence is due to the fact that blowing down forces into the glass hotter and less dense water than is there normally.

**(c) Quantities to be Found Prior to Calculations of Results.**

The proximate analysis of the coal is generally expressed in percentage by weight of the coal "as fired," that is, including moisture. To find the weights per pound of dry coal, it is necessary only to divide each of the items by 100 minus the percentage of moisture. Thus, for the data of the test previously cited,

$$m = 2.75 - (100 - 2.75) = 0.0283 \text{ lb. of moisture;}$$

$$vm = 0.0617 \text{ lb. of volatile matter;}$$

$$fc = 0.807 \text{ lb. of fixed carbon;}$$

$$a = 0.132 \text{ lb. of ash;}$$

$$vm + fc = 0.869 \text{ lb. of combustible.}$$

It is necessary to know the **total carbon per pound of dry coal**. If the ultimate analysis has been made, this figure is available; otherwise, it is necessary to estimate it. For semi-anthracite and bituminous coals, the method quoted under Test 35 (b) may be used, if the proximate analysis only has been made. For anthracite coals in these calculations sufficient accuracy is attained by taking volatile carbon as 1 to 2 per cent of the combustible when the volatile matter is from 5 to 10 per cent. Then,

with the given data, the total weight of carbon in 1 lb. of dry coal is

$$C_t = 0.807 + 0.01 \times 0.869 = 0.816 \text{ lb.}$$

The proportion of **carbon** which is **wasted through cleaning and grate**,  $C_a$ , may be found in two ways. The first method is as follows.

The total carbon in the refuse equals the total amount of refuse minus the ash from the total coal fired. The ash from the total coal fired equals the total quantity of coal times the proportion of ash in it as given by the proximate analysis. Then,

$$C_a = \frac{\text{Total refuse, lbs.} - \text{Total coal, lbs.} \times \text{Per cent ash}}{\text{Total coal, lbs.} - \text{Per cent moisture in coal}},$$

the denominator being the total weight of dry coal.

The second method uses the proximate analysis of the ash.

$$C_a = \frac{\text{Total refuse, lbs.} \times \text{Weight of C in 1 lb. refuse}}{\text{Total coal, lbs.} - \text{Per cent moisture in coal}}.$$

In the first equation it is assumed that all of the ash coming from the coal fired during the main test is removed with the refuse. This, however, is uncertain, so it is better to use the second method. Applying the data of the test under consideration, each pound of the refuse was found to contain 0.20 lb. of carbon. The total coal as fired was 4925 lbs. and it contained 2.75 per cent of moisture. The total weight of refuse was 734 lbs. Then

$$C_a = \frac{734 \times 0.20}{4925 - .0275 \times 4925} = .0307 \text{ lb.}$$

To find the **weight of the carbon burned**.

$$C_g = C_t - C_a.$$

For given data  $C_g = 0.816 - 0.0307 = 0.785$ .

To calculate the factor of evaporation, the heat content of the steam generated is first found. Referring to the test data, page 237, the steam tables give  $H = 305.5 + 0.992 \times 882.5 = 1181$ . Consequently

$$\text{F.e.} = \frac{1181 - (98 - 32)}{970.4} = 1.15.$$

All the items of the heat balance will be figured on the basis of 1 lb. of dry coal. The heat value of the dry coal as found by calorimeter or calculation (see Test 36) will therefore be the heat supplied.

(d) **Equivalent Evaporation, per Pound of Coal as Fired**, is

$$\text{E.e.c.} = \frac{\text{Total water evaporated, lbs.} \times \text{F.e.}}{\text{Total coal as fired, lbs.}}$$

The units of this expression are either pounds of water evaporated from and at 212 degrees per pound of coal, or units of evaporation per pound of coal. For the given data (page 237).

$$\text{E.e.c.} = \frac{36400 \times 1.15}{4925} = 8.5 \text{ lbs. or U.e.}$$

**Per pound of dry coal**

$$\text{E.e.d.c.} = \text{E.e.c.} \div (1 - \text{moisture, lbs.}),$$

For the given data,  $\text{E.e.d.c.} = 8.5 \div (1 - .0275) = 8.65 \text{ lbs. or U.e.}$

**Per pound of combustible**

$$\text{E.e.cb.} = \text{E.e.d.c.} \div (vm + fc),$$

For the given data,  $\text{E.e.cb.} = 8.65 \div 0.869 = 9.95 \text{ lbs. or U.e. per hr.}$

**Per hour**

$$\text{E.e.hr.} = \frac{\text{Total water evaporated, lbs.} \times \text{F.e.}}{\text{Total time, hours.}}$$

For the given data,

$$\text{E.e.hr.} = (36400 \times 1.15) \div 12 = 3440 \text{ lbs. or U.e. per hr.}$$

(e) **Over-all Efficiency.** Since there are E.e.d.c.  $\times 970.4$  B.t.u. added to the feed water for each pound of dry coal fired, and since the heat available from this coal is its heat value, the efficiency is,

$$\frac{\text{E.e.d.c.} \times 970.4}{\text{Heat value of dry coal}}$$

For the given data, over-all efficiency =  $8.65 \times 970.4 \div 13,040 = 65$  per cent.

(f) **Loss to Carbon Wasted in Refuse.** Taking the heat value of carbon as 14,600 B.t.u., this loss per pound of dry coal is

$$14,600 C_a.$$

For the given data,  $14,600 \times 0.0307 = 448$  B.t.u., or  $448 \div 13,040 = 3.4$  per cent.

(g) **Loss of Heat to Dry Exhaust Gases.** This determination involves the measurement of the excess of temperature of the flue gas over that of the boiler room, and of the weight of dry flue gas per pound of dry coal,  $W_a$ . For the specific heat, an average value, 0.24, may be used with sufficient accuracy. Then the loss is

$$0.24 \times W_a \times (T_e - t).$$

To find  $W_a$  the formula given on page 179 may be used. For the given data

$$W_a = \frac{3.04 \times 79.5 \times .785}{11.0 + 0.5} + .785 = 17.3 \text{ lbs. per lb. of dry coal}$$

and the loss is  $0.24 \times 17.3 \times (680 - 80) = 2490$  B.t.u. or  $2490 \div 13,040 = 19.1$  per cent.

(h) **Loss of Heat to Water Vapor in the Flue Gases.** To calculate this, we must know the weight of vapor per pound of dry coal and its heat content above room temperature, including latent heat. The vapor carried in as humidity of the air need not be considered

because it is small in quantity and its latent heat is not added by the coal. This leaves only the vapor from the moisture in the coal and that due to combustion of hydrogen to consider. From the formulas for  $W_v$ , the weight of vapor per pound of coal, on page 180, and for the heat of steam when mixed with flue gases on page 128, we have

$$\text{Loss per pound dry coal} = (9H_t + m)(1090 + 0.46T_s - t).$$

$H_t$ , the weight of hydrogen per pound of dry coal, may be determined according to Test 35 (b) in the case of semi-anthracite and bituminous coals if the ultimate analysis has not been made. For the given test  $H_t = 0.025$ , and the heat lost is  $(9 \times 0.025 + 0.0275)(1090 + 0.46 \times 680 - 80) = 335$  B.t.u. or  $335 \div 13,040 = 2.6$  per cent.

(i) **Loss of Heat Due to Incomplete Combustion.** Considering only the incomplete combustion of carbon, the loss of heat when 1 lb. of carbon burns to CO instead of  $\text{CO}_2$  is  $14,600 - 4400 = 10,200$  B.t.u. (see page 154). In the boiler test there are  $W_c$  lbs. of carbon per pound of dry coal burned to CO. An expression for  $W_c$  is derived on page 180. The loss is

$$10,200 \times W_c \text{ B.t.u. per pound of dry coal.}$$

For the given data,

$$\text{Heat lost} = 10,200 \times \frac{0.5 \times 0.785}{11 + 0.5} = 338 \text{ B.t.u. or } 338 \div 13,040 = 2.7 \text{ per cent.}$$

(j) **Radiation** is found by difference. For the given data, it is,

$$100\% - 65\% - 3.4\% - 19.1\% - 2.6\% - 2.7\% = 7.2\%$$

(k) **Other Quantities.** The boiler horse-power is, as previously defined,

$$\frac{3440}{34.5} = 99.9 \text{ B.h.p.}$$



The equivalent evaporation per hour per square foot of heating surface is,

$$\frac{\text{E.e.hr.}}{\text{Sq.ft. of surface}}$$

For the given boiler, the heating surface was 1000 sq. ft., so E.e.hr. per square foot =  $3440 \div 1000 = 3.44$  lbs. or U.e.

The grate surface of this boiler was 30 sq. ft., so the rate of combustion of dry coal, as previously defined, was  $399 \div 30 = 13.3$  lbs. per hour per square foot.

**Efficiency of the Grate and Cleaning.** The proportion of the combustible lost is  $C_a \div (vm + fc)$ . Consequently this efficiency is,

$$1 - \frac{C_a}{vm + fc}$$

For the given data, this is  $1 - 0.0307 \div 0.869 = 96.5$  per cent.

**Efficiency of the boiler and furnace** (exclusive of grate) is the over-all efficiency divided by the efficiency of the grate. For the given data, this is  $65 \div 96.5 = 67.4$  per cent.

This efficiency is often defined as the heat absorbed per pound of the combustible burned divided by the heat value per pound of combustible.

**The excess coefficient** may be found from the expression derived on page 181.

Besides the total water and coal curves, it is well to plot, after the test, all the readings of pressure, feed water, flue and boiler room temperatures, per cent CO<sub>2</sub> and the draft in inches of water. These will be broken curves. They will aid deductions of causes of any irregularities in the performance.

A few barometer readings are desirable in connection with natural draft.

All circumstances that may affect the conditions of the test should be carefully noted for possible future reference.

**Problem 50<sub>1</sub>.** A 50-H.p. boiler delivers steam at 175 lbs. gage, superheated 100°, from feed water at 50°. What is the F.e.? How many pounds of feed water will be needed per hour during a test? Assuming an efficiency of 70 per cent and anthracite coal with 15 per cent ash, about how many pounds of coal will be needed per hour? *Ans.*, 1.27; 1360 and 190 lbs.

**Problem 50<sub>2</sub>.** For a given natural draft, there is a corresponding flue temperature necessary to maintain it. If the excess air during the operation of a boiler varies, and all other conditions remain constant, deduce the relation between per cent CO<sub>2</sub> and efficiency. Note that efficiency = 1 - a constant - loss to dry exhaust gases, approximately.

## 51. TESTING THE ADJUSTMENT OF AN INTERNAL COMBUSTION ENGINE

**Principles.** Most of the modern internal combustion engines work on the Otto cycle which takes four strokes for a complete series of events. First, the suction stroke, in which the piston,

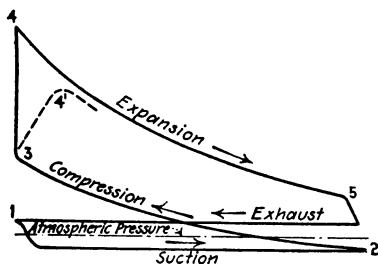


FIG. 88.—Otto Cycle.

advancing from the head end, takes in a charge of gas or vapor fuel mixed with air. Second, the compression stroke, returning, in which the charge is compressed. Third, the expansion stroke, at the beginning of which the mixture is exploded and afterward performs the useful work. Fourth, the exhaust stroke, during

which the burned gases are discharged. Fig. 88 shows this cycle of operation on a  $PV$  diagram. There are two valves to control the charge, one for inlet and one for exhaust. Fig. 89 shows where these valves open and close relative to the crank position and on the indicator diagram. It is seen that the inlet valve opens a little after dead center. This is arranged so that it will open after the exhaust valve has closed. With engines of comparatively low speed, the inlet valve may close a little

after dead center so as to provide a good opening for the charge. The exhaust valve should open a little before dead center and close a little after the opposite dead center to allow free egress of the burned gases, but it should close before the inlet valve opens. The angular distances of the crank from the dead center positions, corresponding to the valve events, depend upon the speed and type of engine. The makers usually furnish this information.

Ignition of the charge should occur in the neighborhood of dead center; generally before, when the engine is running. This is because it takes an appreciable time for the gas mixture to rise

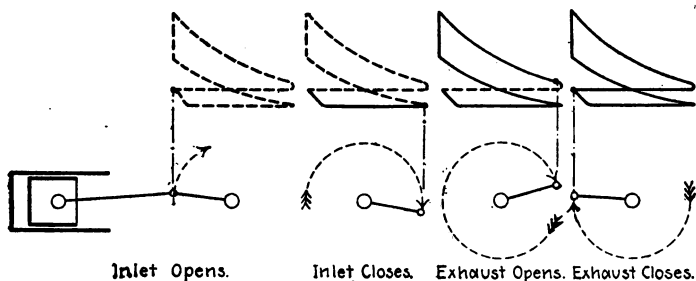


FIG. 89.—Gas Engine Valve Events.

to its maximum pressure after ignition. For maximum power, the greatest pressure should occur at the beginning of the stroke; consequently, ignition should "advance" this position. The greater the rotative speed, the greater will be the angular distance, or advance, between the crank and dead center when ignition takes place. Some fuels burn more slowly than others; for such the ignition must be more advanced. The angular distance varies between zero and 50 degrees.

(a) **Timing the Valves.** This is done by reference to the crank positions when the valves are opening or closing. A prick point should be made on the flywheel rim to mark the line of the crank,

and the position of this prick point located by trammels or otherwise when the engine is on dead center. The angular distance between the prick point and its dead center position will determine the motion of the crank for any valve event.

The valves usually are operated by cams on a cam shaft driven by 2 to 1 gears, *a* and *b*, from the main shaft. See Fig.

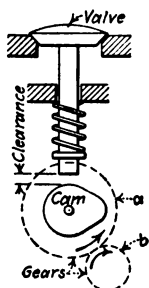


FIG. 90.  
Gas Engine Valve.

90. There is a certain amount of clearance between the valve stem and the cam when the latter is not acting. By changing this clearance the timing of the valve will be altered. From the figure it will be seen that to make the clearance greater makes the valve open later and close earlier. The timing may be changed also by changing the relative positions of the gears *a* and *b*. Thus, in the figure, if gear *a* is turned back (clockwise) so that it meshes with one tooth behind on *b*, both opening and closing will be later.

There is usually some provision made for changing the clearance. The procedure in timing the valves, then, is first to locate the crank positions corresponding to opening and closing, and then to correct, if necessary, by changing the clearance or gear mesh or both. The point of opening or closing may be accurately fixed by turning the valve stem with the fingers; the friction of the valve when seated is easily felt. It should be noted that the valve events are somewhat different when running because of the expansion of the valve stems due to the working temperature. To allow for this, the engine may be operated until it warms up, and then timed. The makers' instructions for timing are usually for the cold condition.

(b) **Timing the Ignition.** For jump spark ignition with batteries or low tension magneto, this is readily done by removing the spark plug and noting the crank position to correspond

with the occurrence of the spark. For high tension magneto systems, the timer on the magneto must be observed. For make and break systems, it is easy to note the crank position when the igniter breaks contact, as it is visible.

For economical running, ignition should be advanced as far as possible without causing pounding or diminution in speed.

**(c) Adjustment of Mixture.** Whether the engine uses oil vapor or fuel gas, there is one device for setting the amount of air supplied and another, the amount of fuel. Strictly speaking these should be readjusted for every load on the engine, but usually a single adjustment is made to give the best mixture at usual running loads. As a general rule, it may be accepted that the most economical mixture is as dilute a one as may be had without causing misfiring or objectionable slowing down of the engine. Also, for economy, the fuel valve should be open as little as possible. This is especially the case with engines with throttling governors. If the engine is given too much fuel, it tends to speed up which puts upon the governor the work of throttling it so that the suction in the cylinder is greater and the line 1-2, Fig. 88, is lower. The compression then starts with a lower pressure at 2 and ends with a lower pressure at 3. As the thermal efficiency is lower, the less is the compression pressure, it is seen that throttling diminishes efficiency. Further, the greater the throttling, the larger will be the lower loop of the indicator diagram. This represents the work lost in pumping the charge into the cylinder.

With an engine regulated by a governor, the procedure is as follows. The desired load is applied by brake or otherwise and the valve controlling the flow of fuel is set with a slightly larger opening than is thought to be correct. The air opening is then increased until the engine shows signs of slowing down, as determined by a tachometer. Records of speed and valve settings for this trial are kept. The procedure is then repeated with a slightly less fuel valve opening. The smallest fuel valve

opening with the largest air valve opening which will not cause objectionable slowing down or irregular firing may be accepted as the best setting for the applied load. In this connection it is essential that the exhaust be observed during each trial; it should be clear and regular.

Variable speed engines supplied with gasoline mixed in carburetors, such as automobile and marine engines, are more difficult to adjust than constant speed engines, but the same principles are to be observed. Many carburetors are made with automatic adjustment to accommodate changes of speed, and the adjustment is made with the engine delivering no torque. The proper setting must be judged mainly by the appearance and sound of the exhaust.

**(d) Test of Timing by Indicator.** This applies to engines running at not more than 250 R.p.m. if the ordinary indicator is to be used. Special gas engine indicators are made to handle about double this speed. Higher than this, only optical indicators may be used with success. A light spring should be used in the indicator, about 20 lbs., and the indicator piston arranged with a stop so that it will not register much above atmospheric pressure, in order to get the suction diagram on a larger scale.

If the inlet valve opens too late, the line from 1, Fig. 88, will not have as sharp a drop as it should and the suction line will be lower. If the inlet valve closes too late, the compression curve will not start sharply from the point 2. If the exhaust valve opens too late, expansion is carried a little beyond point 5, but the beginning of the exhaust line will be higher on account of the greater back pressure. If it closes too late, the effect is much the same as for too late opening of the inlet. When the valve events are too early, opposite effects will be recorded.

The indicator diagram does not show fine differences in the timing of the valves because the events occur near the ends of the stroke where the indicator drum has but little motion corresponding to the angular motion of the crank. For this reason,

it is useful to set the reducing motion of the indicator 90 degrees ahead of the crank, so that the valve events will be shown in the middle of the indicator diagram.

If the ignition is properly advanced, the line 3-4, Fig. 88, will be vertical. If ignition is too late, this line will slant as shown by the dotted line 3-4', causing a diminution of power; if too early it will slant in the opposite direction with the same effect, accompanied by pounding.

**Problem 51<sub>1</sub>.** An inlet valve is found to open too early and close too late. The exhaust valve on the same cylinder opens and closes too early. What should be done to correct the timing, if the two cams are integral with the cam shaft?

**Problem 51<sub>2</sub>.** Draw two indicator diagrams, each superposed on a normal diagram like Fig. 88, to show the effects of too early and too late valve events.

**Problem 52<sub>3</sub>.** Draw a suction diagram to be expected with normal setting when the reducing motion is set  $90^\circ$  ahead of the crank. Draw the work diagram.

## 52. MECHANICAL EFFICIENCY TEST OF AN INTERNAL COMBUSTION ENGINE

**Principles.** Exactly the same methods and formulas apply to this test as to Test 41, with the exception of the determination of  $N$  in the formula for indicated horse-power. Internal combustion engines are generally single acting, so, with throttling governors and regular firing, there is only one power stroke for each two revolutions of a crank. Under these conditions, the value of  $N$  is one-half the revolutions per minute. When the engine governs on the hit-and-miss principle, however, the actual number of explosions per minute must be counted. This also applies with a throttling engine if the firing is irregular.

The indicator diagram, Fig. 88, contains a negative area which must be subtracted from the upper area to get the net

mean effective pressure. This may be done with the planimeter by traversing the upper area in a clockwise direction, and the lower, anti-clockwise. The indicated horse-power obtained from the result is the "net" I.h.p. which is greater than the brake horse-power by the horse-power given to mechanical friction. The lower loop of the diagram gives the work lost to fluid friction, that is, to pumping the charge into the cylinder. It may be determined accurately only by using a light indicator spring as explained under Test 51 (d).

(a) **Determination of Indicated Horse-power.** See Test 41 (a). If the engine governs on the hit-and-miss principle, the explosions may be counted by timing the sound of the exhaust if the speed is not too fast. Special devices are made for this purpose when a continuous record is wanted. One may be made by tapping a small pipe into the exhaust and covering it with a flap held down by a spring. An explosion causes the flap to rise and so actuate the lever of a revolution counter.

(b) **Friction Horse-power.** The mechanical friction is obtained as for Test 41 (b). The fluid friction may be expressed as horse-power calculated from the mean-effective pressure of the lower loop, the value of  $N$  being the same as for the net I.h.p.

(c) **Efficiency.** (d) **Speed Regulation.** See Test 41 (c) and (d).

**Problem 52.** A 2-cylinder, 6-in.  $\times$  8-in.  $\times$  350 R.p.m. gas engine, governing on the hit-and-miss principle, misses one out of every five explosion strokes. If the net mean-effective pressure is 20 and 21 lbs. in the two cylinders, what is the I.h.p.?

*Ans.*, 3.28 H.p.

**Problem 52.** What is the engine constant of a 6-in.  $\times$  8-in. gas engine? If the brake horse-power is 5, and the M.e.p. of the upper loop of the diagram is 70, and of the lower loop 5 lbs., and the engine fires 163 times per minute, what is the horse-power lost to mechanical friction? To fluid friction?

*Ans.*, .000571, 1 H.p.. .077 H.p.

**Problem 53.** Compare by actual test the speed regulation of an engine working on the hit-and-miss principle with a throttling engine. Determine coefficient of regulation at full and half loads.



## 53. ECONOMY TEST OF A GAS ENGINE

**Principles.** A complete economy test of a gas engine should include, besides the fuel consumption, the determination of the various losses so that their distribution may be studied. All of the heat energy of the fuel supplied must be accounted for in the form of useful work and losses. The equation between these quantities is called the *heat balance*.

The losses involved in the operation of a gas engine are as follows. Mechanical friction, heat carried away by jacket water, by the dry exhaust gases, by steam in the exhaust, by unburned fuel gas, and stray losses as radiation, etc. It is convenient and logical to base the calculation of these losses upon 1 cu. ft. of fuel gas under standard conditions of temperature and pressure. The heat balance may then be written, "heat supplied by 1 cu. ft. of gas equals the heat from 1 cu. ft. turned into useful work plus heat from 1 cu. ft. lost to friction plus, etc."

To measure the useful work and friction and jacket losses on this basis it is necessary to meter the fuel supplied in a given time. The other quantities of the heat balance require the analyses of both the fuel and exhaust gases. Generally, for the fuel gas analysis, a chemist's services are needed. Strictly speaking, a fair sample of the gas used during the test should be analyzed, but when the fuel is such as city illuminating gas which does not vary much from day to day, an average analysis is fairly representative and may be used without considerable error.

The table, page 149, gives the average composition of the principal gas engine fuels.

The exhaust gas analysis may be made with the Orsat apparatus. The chief constituents are  $\text{CO}_2$ ,  $\text{O}_2$ , and  $\text{N}_2$ , but an analysis for  $\text{CO}$  should not be omitted. A determination of  $\text{H}_2$  is desirable, but not necessary if the  $\text{CO}$  is less than 1 per

cent, as it is in proper operation. Since the CO is the least readily burned of the fuel gas constituents, it is a fair assumption that the hydrogen and hydrocarbons are completely burned if the CO is nearly all burned, and experiment bears out this assumption.

Knowing the fuel and exhaust gas analyses, the weights of gases in the exhaust pipe per cubic foot of fuel supplied may be calculated from the formulas deduced under Test 38 (b). This division should be re-read and thoroughly understood.

**Sampling.** For a chemist's analysis of the fuel gas, it is best to take a continuous sample covering the whole time of the test. This can be done by connecting a 5-gal. flask with rubber tubing to the gas main and syphoning water from the flask at a slow and uniform rate thus drawing in the sample. Sampling for heat value determination may be made as described on page 160, Test 37.

For throttling engines, the exhaust gas may be sampled as explained on page 173, Test 38. Hit-and-miss engines, it should be remembered, occasionally discharge air into the exhaust pipe which has not mixed with fuel. This air carries a certain amount of heat, though less than that carried by burned gases, and for this reason should be counted separately. It is possible to make an arrangement in the exhaust pipe by which the discharge from a missing stroke is by-passed and its temperature and quantity measured separately from the main exhaust. As this is a difficult matter, results may be obtained by averaging in the air from a missing stroke with the working exhaust, the sample then being taken as for a throttling engine. A thermometer in the exhaust pipe will show a mechanical average of the variable temperature which may be used for calculating the heat lost. When the exhaust from a hit-and-miss engine is sampled in this way, the excess coefficient cannot be figured from the analysis without first applying a correction for the air taken in by the engine, but not used to mix with the fuel.

Since both the fuel and exhaust gases are generally under pressure, it is easier and more accurate to obtain samples than in boiler work.

**Duration of Test.** This should be several hours, preferably, but, if on a small engine, a test of only one hour long will give valid results if the engine is previously run for fifteen or twenty minutes under the conditions to be maintained during the test. The readings to be mentioned later should be taken sufficiently often to obtain a fair average. The useful horse-power should be maintained at a constant value and all other conditions kept as uniform as possible. The test results are not valid unless the gas consumption is uniform as shown by meter readings taken at equal time intervals.

To make the methods clear, the measurements from a gas engine test will be worked through for the various results. Table No. 1 gives these measurements, and also the notation used in the formulas. The fuel used was illuminating gas manufactured in Syracuse, an average analysis of which is given in columns (1) and (2) of Table No. 2, page 257.

**(a) Fuel Consumption, Cubic Feet of Gas per Brake horse-power-hour.** Since the weight of fuel in a cubic foot varies with the pressure and temperature, the gas meter readings should be referred to standard conditions, namely, a pressure of 29.92 ins. of mercury and a temperature of 32° F. The following formula may be used. (See Table No. 1 for notation.)

$$V' = 16.4 \frac{B + \frac{P}{13.6}}{T_r + 460} \times V.$$

$P$  and  $T_r$  necessitate the use of a water manometer and a thermometer at the point in the gas main where the gas meter is placed. For small size engines, the meter may be the usual gas meter, or a gasometer, but for large sizes, meters of the pitot, venturi, or orifice type may be used. When the engine is so large as to

TABLE 1

# GIVING DATA FROM A GAS ENGINE TEST AND NOTATION USED IN FORMULAS

B.h.p. = brake horse-power	=25
I.h.p. = net indicated horse-power	=30.1
F.h.p. = friction horse-power	
V = number of cubic feet of fuel gas per hour, by meter	=414
V' = ditto, corrected to absolute pressure of 29.92 ins. of mercury and 32° F.	
B = Barometer reading, inches of mercury	29.5
P = pressure of fuel gas above atmosphere, at meter, inches of water	4
F = fuel consumption, standard cubic feet of gas per brake horse-power-hour.	
T <sub>f</sub> = temperature of fuel gas at meter, degrees F.	=68
t <sub>j</sub> = temperature of ingoing jacket water, degrees F.	=61
T <sub>j</sub> = temperature of outgoing jacket water, degrees F.	=133
T <sub>e</sub> = temperature of exhaust gases, degrees F.	=750
t = temperature of air near engine, degrees F.	=70
W <sub>j</sub> = weight of jacket water, pounds per hour	=1210
V <sub>a</sub> = volume of dry exhaust gas from the combustion of 1 cu. ft. of fuel, cubic feet per cubic foot.	
W <sub>a</sub> = weight of dry exhaust gas from the combustion of 1 cu. ft. of fuel, pounds per standard cubic foot.	
W <sub>e</sub> = weight of water vapor, ditto.	
R = ratio by volume of air consumed to fuel gas.	
X = excess coefficient.	
c, h, g = from sums of columns (3), (4), (5), Table 3, respectively.	

## VOLUME ANALYSIS OF EXHAUST GAS, PER CENTS

D = carbon dioxide, CO <sub>2</sub>	=9.0
O = oxygen, O <sub>2</sub>	=8.8
M = carbon monoxide, CO	=0.5
H = hydrogen, H <sub>2</sub>	Not analyzed.
N = nitrogen, N <sub>2</sub>	=81.7

be inconvenient to brake, the useful horse-power may be approximated by subtracting from the indicated horse-power at the running load its value when the engine is running entirely free.

Applying the data of Table No. 1, we have,

$$V' = 16.4 \times \frac{29.5 + \frac{4}{13.6}}{68 + 460} \times 414 = 385 \text{ cu. ft. per hour.}$$

TABLE 2

ANALYSIS OF FUEL GAS USED DURING TEST, AND COMPUTATION OF QUANTITIES USED IN FIGURING RESULTS

(1) Constituents of Fuel.	(2) Per Cent by Volume.	(3) Subscript of $C \times (2)$ .	(4) Ditto of $H$ .	(5) Ditto of $O$ .	(6) Higher Heat Value of (1)	(7) $\frac{(6) \times (2)}{100}$
CO	23.5	23.5	.....	23.5	342	80.5
H <sub>2</sub>	36.4	.....	72.8	.....	348	127
CH <sub>4</sub>	15.3	15.3	61.2	.....	1065	163
C <sub>2</sub> H <sub>4</sub>	6.9	13.8	27.6	.....	1680	116
C <sub>6</sub> H <sub>6</sub>	3.9	23.4	23.4	.....	4000	156
O <sub>2</sub>	1.4	.....	.....	2.8	.....	.....
CO <sub>2</sub>	4.2	4.2	.....	8.4	.....	.....
N <sub>2</sub>	8.4	.....	.....	.....	.....	.....
Sum		80.2 = c	185.0 = 2h h = 92.5	34.7 = 2g g = 17.35		643 = higher heat value of fuel

From which the fuel consumption is

$$F = \frac{385}{25} = 15.4 \text{ standard cubic feet per brake horse-power-hour.}$$

(b) **Heat Supplied.** Since the heat balance is based upon one cubic foot of the fuel gas, the heat supplied is its heating value. It is preferable to use a calorimeter for this determination, such as the Junker, but if one is not available a working value may be obtained by calculation from the fuel gas analysis, Test 37 (c). The calculation is made in column 7 of Table 2.

The gas engine code of the A.S.M.E. specifies that the "higher" heat value be used, that is, the heat obtainable from the fuel including the latent heat of vaporization of the water formed by combustion of the hydrogen. Some authorities contend that the lower heat value is more logical to use in this connection since the gas engine cannot avail itself of this

latent heat, but for the sake of uniformity the recommendations of the code will be followed.

(c) **Heat Converted into Useful Work.** The heat equivalent of one brake horse-power-hour is 2545 B.t.u. Since the fuel consumed to produce this power is  $F$  cu.ft., the heat equivalent of the useful power per cubic foot of fuel is

$$\frac{2545}{F}.$$

Applying the data of the example this is,

$$\frac{2545}{15.4} = 165 \text{ B.t.u.}$$

Dividing this by the heat value gives 25.6 per cent. This is the thermal efficiency.

(d) **Heat Lost to Mechanical Friction.** In power units,

$$\text{F.h.p.} = \text{I.h.p.} - \text{B.h.p.}$$

To base this upon the heat of one cubic foot of fuel, a formula derived the same as the one last given is used, namely

$$\text{Friction loss, B.t.u.} = \frac{2545 \text{ F.h.p.}}{V'}.$$

Applying the data of the example, this is

$$\frac{2545(30.1 - 25)}{385} = 33.7 \text{ B.t.u.}$$

or  $33.7 \div 643 = 5.2 \text{ per cent.}$

(e) **Heat Lost to Jacket Water.** This is the heat absorbed in a given time divided by the fuel used in the same time. Considering hourly quantities,

$$\text{Jacket loss} = \frac{W_j(T_j - t_j)}{V'}.$$

The weight  $W$ , may be measured by a barrel or tank on a platform scales. The temperatures should be read with two thermometers; one in the discharge tank (provided the piping is arranged to discharge without material radiation loss), the other in a thermometer well in the inlet pipe or in a receptacle drawing water from a tap in the main situated similarly to the jacket as to temperature.

Applying the data of the example,

$$\text{Jacket loss} = \frac{1210(133 - 61)}{385} = 226 \text{ B.t.u.}$$

or

$$226 \div 643 = 35.2 \text{ per cent.}$$

**(f) Volume of Dry Exhaust Gas per Cubic Foot of Fuel Gas.**

This quantity is to be used in the various calculations from the exhaust gas analysis. Under Test 38 (b) there is deduced the formula,

$$V_d = \frac{c}{D + M},$$

$c$  is found as in column 3 of Table 2. The subscript of the  $C$  in each fuel gas constituent shown in column 1 is multiplied by the volume percentage of that constituent as shown by column 2. Column 3 gives these products, the sum of which is the value of  $c$ .

For the data of the example

$$V_d = \frac{80.2}{9.0 + 0.5} = 8.44 \text{ cu. ft.}$$

**(g) Loss of Heat to Dry Exhaust Gases.** This determination involves the measurement of the temperature rise of the fuel and air mixing with it, and of the weight of the dry exhaust gases. For the specific heat, an average value, 0.245, may be used with sufficient accuracy. Then

$$\text{Loss to dry exhaust gases} = 0.245 W_d (T_e - t).$$

Under Test 38 (b) there is deduced the formula,

$$W_a = \frac{V_a}{9000} \left\{ 11D + 8O - 7(M + N) \right\},$$

by which the weight of the dry exhaust gas may be found from its analysis.

The temperature in the exhaust pipe should be taken by a high reading thermometer (1000° F.) inserted in a well filled with fine, dry sand, located as near the engine as possible.

Applying the data of the example,

$$W_a = \frac{8.44 \{ 11 \times 9 + 8 \times 8.8 + 7 \times (0.5 + 81.7) \}}{9000},$$

$$= 0.703 \text{ lbs. per cu. ft. of fuel.}$$

$$\text{and loss to dry exhaust gas} = 0.245 \times 0.703 \times (750 - 70)$$

$$= 118 \text{ B.t.u.}$$

$$\text{or} \quad 118 \div 643 = 18.3 \text{ per cent.}$$

(h) **Loss of Heat to Water Vapor in the Exhaust.** Since the higher heat value of the fuel is used, the latent heat of the water vapor is to be included. The small amount of water vapor brought in as humidity of the air and fuel may be ignored, particularly as its latent heat is not added by the fuel. This leaves only the water from the combustion of hydrogen. A formula for the weight of vapor so formed from one cubic foot of fuel gas is deduced under Test 38 (b). Combining this with the expression for total heat of steam under the given conditions (see page 128), we have,

Loss of heat to water vapor =  $W_v \times (\text{total heat of the vapor above room temperature})$

$$= 0.0005h (1090 + 0.46T_v - t).$$



The value of  $h$  is found similarly to  $c$ , as shown by column 4, Table 2.

Applying the data of the example.

$$\begin{aligned}\text{Loss to water vapor} &= 0.0005 \times 92.5 \times (1090 + 0.46 \times 750 - 70) \\ &= 64.3 \text{ B.t.u.}\end{aligned}$$

$$\text{or} \quad 64.3 \div 643 = 10 \text{ per cent.}$$

(i) **Heat Lost by Unburned Fuel Gas.** The volume of unburned CO per cubic foot of fuel gas, according to Test 38 (b), is  $V_4 = V_d M \div 100$ . As the heat value of CO is 342 B.t.u. per cubic foot, the

$$\text{Loss to unburned CO} = 3.42 V_d M.$$

Similarly, if  $H$  per cent of hydrogen is found in the exhaust,

$$\text{Loss to unburned } H_2 = 3.46 V_d H,$$

the higher heat value of hydrogen being 346 B.t.u.

These two expressions may be combined by assuming an average heat value, 344, to apply to both CO and  $H_2$ . Then

$$\text{Loss to unburned CO and } H_2 = 3.44 V_d (M + H).$$

If hydrogen has not been analyzed for,  $H$  may be omitted.

Applying the data of the example,

$$\text{Loss to unburned fuel} = 3.44 \times 8.44 \times 0.5 = 14.6 \text{ B.t.u.}$$

$$\text{or} \quad 14.6 \div 643 = 2.3 \text{ per cent.}$$

(j) **Heat Lost to Radiation, etc.,** is found by subtracting from the heat value of the fuel the other quantities of the heat balance (c) to (j), or by subtracting from 100 per cent these quantities expressed in per cents. The unaccounted for heat includes that used for pumping the fuel mixture into the cylinder if the area of the lower loop has been subtracted from the upper of the indicator diagrams. This quantity could be separately measured,

but it is generally sufficient to include it in the friction loss as is done when the lower loop is altogether ignored, or to list it under radiation.

Applying the data of the example,

$$\begin{aligned}\text{Loss to radiation, etc.} &= 643 - 165 - 33.7 - 226 - 118 - 64.3 - 14.6 \\ &= 21.4 \text{ B.t.u.,}\end{aligned}$$

$$\text{or } 21.4 \div 643 = 3.4 \text{ per cent.}$$

(k) **Cubic Feet of Air per Cubic Foot of Fuel Gas.** This may be figured from the formula deduced under Test 38 (b),

$$R = \{V_a(D + O + 0.5M) + 0.5h - g\} \div 21$$

in which  $g$  is found under column 5, Table 2.

Applying the data of the example,

$$\begin{aligned}R &= \{8.44 \times (9.0 + 8.8 + 0.5 \times 0.5) + 0.5 \times 92.5 - 17.35\} \div 21 \\ &= 8.63 \text{ cu. ft.}\end{aligned}$$

(l) **The Excess Coefficient.** The following formula may be used.

$$X = \frac{21R}{c + 0.5h - g}$$

Applying the data of the example,

$$X = \frac{21 \times 8.63}{80.2 + 0.5 \times 92.5 - 17.35} = 1.66.$$

**Problem 53.** The heat balance is found for a number of runs at different loads, and the items plotted thus: Eff. vs. B.h.p., Eff. + Friction vs. B.h.p., Eff. + Friction + Jacket loss vs. B.h.p., etc. Sketch the resulting curves in what you would expect to be their form and give reasoning.

**Problem 53.** If the air used in the test example given above was saturated with humidity, what would be the per cent loss to the heat carried away by this vapor? Ans., 0.52%.

**Problem 53.** If the engine of the test example had been of the hit-and-miss type, and during the test missed one out of five explosive strokes, all other data being the same, what would have been the excess coefficient?

Digitized by Google Ans., 1.33.

#### 54. ECONOMY TEST OF AN AUTOMOBILE ENGINE

**Principles.** Engines of this type are variable speed. A complete test will therefore consist of enough runs at different speeds to cover the working range. As the rotative speeds are high, these engines cannot be indicated except with the optical indicator. The exhaust gas analysis cannot always be relied upon for the determination of the losses, because irregular firing, incomplete combustion, or lubricating oil in the exhaust will make erroneous the percentages as usually found. If conditions are such that the exhaust gas analysis is valid, the methods of the preceding test and the formulas for coal combustion, Test 38 (b), are applicable. In these formulas,  $C_g = C_f$  = weight, in pounds, of the carbon in the fuel per pound of fuel; since all the carbon is gasified. In the formula for the weight of water vapor in the exhaust, the value of  $m$ , moisture in the fuel, drops out. The results from the formulas will be based on 1 lb. of fuel.

American gasoline, specific gravity about 0.7, contains roughly 83 per cent of carbon, 15 per cent of hydrogen, and less than 1 per cent of oxygen, by weight. Kerosene is a little higher in carbon and lower in hydrogen. The higher heat value of gasoline is about 20,000 B.t.u.; the lower, about 19,000 B.t.u. per pound. The heat values of kerosene per pound are practically the same, but based on the gallon they are higher, the specific gravity of kerosene being 0.8 or more.

(a) **Determination of Torque and Brake Horse-power at Various Speeds.** The ordinary prony brake is difficult to use when it is sought to maintain a uniform load long enough for a fuel consumption test, because of the lack of uniformity of the frictional resistance. Fan brakes have been much used for the purpose, but they are not strictly reliable. Probably the best results ensue from the use of a well designed water brake. The writer has used a rope brake with considerable success.

At each speed the carburetor and spark should be adjusted as under regular operation of the engine, so as to get the maximum brake thrust without using an excess of fuel. If the fuel consumption is not measured, each run should be at least fifteen minutes in duration after uniform conditions have been established.

It is useful to plot both torque and brake horse-power against speed in revolutions per minute.

The brake horse-power of automobile engines is often estimated by empirical formulas, especially for the purpose of rating. The Association of Licensed Automobile Manufacturers uses the formula

$$\text{B.h.p.} = D^2 \times \text{number of cylinders} \div 2.5,$$

$D$  being the bore in inches. This is referred to as the A.L.A.M. rating and applies to four-cycle engines at an assumed piston speed of 1000 feet per minute. It generally gives too high results.

Under favorable conditions the mean-effective pressure from the combustion of gasoline in the modern four-cycle engine is about 80 lbs. per square inch. Using this value and assuming a mechanical efficiency, a value for the brake horse-power may be calculated with which to compare test results. The mechanical efficiency may be estimated as follows. Compression is relieved from the cylinders by removing spark plugs, and the engine then caused to revolve as rapidly as convenient by pulling on a rope wrapped around the flywheel. The pull may be measured with a spring balance. From this measurement the torque necessary to overcome friction may be calculated. Assuming that the frictional torque remains constant, the mechanical efficiency equals

Brake torque at given speed

---

Brake torque at given speed  $\div$  torque to friction

(b) **Fuel Consumption, Gallons or Pounds per Brake Horsepower-hour.** The general principles of this determination are the same as for other fuel consumption tests except as regards the measurement of fuel. For this purpose a calibrated tank may be used, containing the oil with which to supply the engine. In some cases, it is desirable to keep the head of oil on the float valve of the carburetor practically constant. An arrangement may then be used similar, on a small scale, to that for measuring feed water to a boiler, Fig. 87. Another plan is to place the supply vessel on a scales and to syphon the fuel from it.

**Problem 54.** A 3-cylinder, 4-in.  $\times$  4 in., 4-cycle engine develops 60 lb.-ft. torque at 800 R.p.m. The friction torque is 15 lb.-ft. What are the B.h.p., the probable mechanical efficiency and M.e.p.? Compare the B.h.p. with the A.L.A.M. rating, allowing for speed. *Ans., M.e.p. = 37.5.*

## 55. ECONOMY TEST OF A GAS PRODUCER

**Principles.** The distribution of the heat of the coal consumed by a producer is in four directions. First, the useful heat, appearing as the calorific value of the dry, cool, gas; second, the heat carried by the gas as sensible heat and as latent heat of the excess steam; third, the heat lost to good fuel removed with ash; fourth, the heat lost through deposits of tar and soot, absorption and leakage of gases, and radiation from the producer.

The selection of methods of measurement of these heat quantities is governed somewhat by the type of producer and fuel, whether pressure or suction, anthracite or bituminous. In any case, it is necessary to measure the coal and ash during a test of sufficient duration, and to provide a proximate analysis of the coal and a chemist's analysis of an average sample of the gas. The gas usually contains between 10 and 20 per cent of hydrogen, 1 to 3 per cent of marsh gas, and a small amount of olefiant gas, in addition to  $\text{CO}_2$ ,  $\text{CO}$ , and  $\text{N}_2$ . These make it not easy

to analyze, although it may be done without much labor if the apparatus is available.

The calorific values of both coal and gas may be calculated from their analyses.

The volume of gas generated may be figured from the gas analysis with acceptable accuracy in most cases, thus dispensing with the bothersome meter question. This calculation depends upon the assumption that all the carbon gasified from the fuel appears in the gas analysis. It is therefore approximate since the condensible hydrocarbons distilled from the coal throw down some carbon in the form of tar, and some disappears as soot. In the case of anthracite gas, or well fixed bituminous, this is negligibly small since, generally, it does not involve more error than is inherent to the coal measurement.

The largest error likely to occur is in the measurement of the coal consumed, in that the producer may contain a very different amount at the end of the test from its contents at the beginning. To avoid this, the conditions of the producer at **starting and stopping** should be noted and made in all respects as nearly the same as is possible. Starting and stopping should be timed just after a regular cleaning. **The duration of the test** should be long enough to reduce the probable error caused by the variation in fuel content of the producer to a small percentage of the total coal fired, which, in general, will not be less than twelve hours. If the weight of each charge of coal is plotted as a total against the time at which it is consumed, the resulting curve will show by its straightness the uniformity of coal consumption. For such uniformity all external conditions must be kept as constant as possible and the producer should be run for some hours under the required conditions before the main test is started.

The ash and refuse should be collected at regular cleaning intervals and weighed after drying. It is best to secure a sample of the total refuse just before weighing and to determine its

moisture and carbon as in the proximate analysis of the coal. It is often awkward to dry all the refuse thoroughly, but, if the percentage of moisture is known, the weight of dry refuse may be readily figured.

**Sampling** of the coal and ash should be done as for a boiler test. Sampling of the gas should be the same as for the fuel gas during a gas engine test (see page 160). The gas should be collected before entering the scrubber, and after leaving it if it is desired to find the effect of absorption or leakage.

The weight of steam furnished to a suction producer exclusive of that due to vapor in the air and moisture in the coal may be found by supplying the vaporizer with water from a calibrated vessel and collecting and weighing the overflow. For a pressure producer, an orifice or nozzle method may be used to meter the steam, or it may be measured by weighing the feed water evaporated by the auxiliary boiler.

The calculations from the test of an anthracite suction producer plant will first be given and then the modifications necessary for other types. Tables 1 and 2, pages 268 and 269, give the required test data together with the notation used in the formulas. The data are part of the full observations of a test reported in the Journal of the A.S.M.E., December, 1909, for which the ultimate as well as the proximate coal analysis and calorimetric determinations of fuel and gas were made, and the gas metered. The reader may thus compare the results by the approximate methods to be presented with the more exact results of the report.\*

All the items of the heat balance will be figured on the basis of 1 lb. of dry coal. The calorific value of the dry coal will thus

\* Note that these results are given for the standard gas condition of 62° F. and 30 ins. of mercury, whereas the standard here used is for 32° F. and 29.92 ins. The volume per pound of the latter is 0.943 times that of the former. Also the specific heats of the gases are different, the values here quoted being by later authorities.

TABLE 1

# GIVING DATA FROM A SUCTION GAS PRODUCER TEST AND NOTATION USED IN FORMULAS

Total weight of coal charged, pounds = 798

Total weight of ash and refuse, pounds = 85

Total weight of water used in vaporizer = 268

## PROXIMATE ANALYSIS OF THE COAL. PER CENTS BY WEIGHT OF THE COAL FIRED

Moisture = 2.75

Volatile matter = 6.00

Fixed carbon = 78.45

Ash = 12.8

$m, vm, fc$  = weights of moisture, volatile matter, and fixed carbon per pound of dry coal, respectively.

Weight of carbon in 1 lb. of dry refuse, pounds = 0.388

$T_p$  = temperature of gas leaving producer, degrees F. = 1108

$t$  = temperature of fire room, degrees F. = 82

$C_t$  = total weight of carbon in 1 lb. of dry coal, pounds.

$C_a$  = weight of carbon wasted in ash, per pound dry coal, pounds.

$C_g$  = weight of carbon gasified per pound dry coal, pounds.

$c, p$  = sums of columns 3 and 5, Table 2, respectively.

$V$  = volume of dry gas generated per pound of dry coal, standard cubic feet.

$W_p$  = weight of the producer gas per pound of dry coal, pounds.

$W_a$  = weight of the air furnished per pound of dry coal, pounds.

$W_s$  = weight of steam furnished per pound of dry coal, from the vaporizer, pounds.

$ws$  = weight of steam not dissociated per pound of dry coal, pounds.

be the heat supplied; the sensible heat carried in by air, water, etc., being neglected.

The principles upon which most of the calculations depend are given under Test 38 (b), coal combustion, and should be thoroughly understood.

## (a) Quantities to be Found Prior to Calculation of the Results.

The proximate analysis of the coal should be based upon its dry weight. This calculation is the same as for Test 50 (c), as is that for  $C_t$ , the total carbon per pound of dry coal; since the same



TABLE 2

ANALYSIS OF GAS GENERATED DURING TEST, AND COMPUTATION OF QUANTITIES USED IN FIGURING RESULTS

(1) Constituents of Gas.	(2) Percentage by Volume.	(3) Subscript of C $\times$ (2)	(4) Molecular Wt. of (1).	(5) (2) $\times$ (4)	(6) Higher Heat Value of (1).	(7) $\frac{(6) \times (2)}{100}$
CO	27.0	27.0	28	756	342	92.4
H <sub>2</sub>	10.4	.....	2	20.8	348	36.2
CH <sub>4</sub>	1.8	1.8	16	28.8	1065	19.2
C <sub>2</sub> H <sub>4</sub>	0.0	0	28	0	1680	0
O <sub>2</sub>	0.2	.....	32	6.4	....	
CO <sub>2</sub>	4.2	4.2	44	184.8	....	
N <sub>2</sub>	56.4	....	28	1579.2		
		$c = 33.0$		$p = 2576.0$		147.8 = higher heat value of gas

fuel analysis has been assumed. The methods of finding  $C_a$  and  $C_g$  are also the same as given under Test 50 (c). For the data given by Table 1,

$$\begin{aligned}\text{Total weight of dry coal consumed} &= 798 - 0.0275 \times 798 \\ &= 776 \text{ lbs.}\end{aligned}$$

$$\text{and } C_a = \frac{85 \times 0.388}{776} = 0.042 \text{ lb.}$$

The carbon gasified is

$$C_g = C_t - C_a = 0.816 - 0.042 = 0.774 \text{ lb. per pound of dry coal.}$$

The volume of dry gas generated per pound of dry coal, under standard conditions of pressure and temperature, may be found from a formula derived similarly to that for  $V_a$ , page 178. For producer work  $D+M$  becomes  $c$  since carbon may

exist in other gases than  $\text{CO}_2$  and  $\text{CO}$ .  $c$  is found as shown in column 3 of Table 2. Then

$$V = \frac{2980 C_g}{c}.$$

For the given data,

$$V = \frac{2980 \times 0.774}{33} = 69.9 \text{ cu. ft. per lb. dry coal.}$$

The weight of dry gas per pound of dry coal may be obtained from its volume if the weight of a cubic foot of the gas is known. For the latter quantity, the method of finding the density of a gas described on page 177 may be used. The weight of a molecule-foot of the producer gas is  $p \div 100$ ,  $p$  being determined as in column 5, Table 2. The weight of a molecule-foot of hydrogen is 2, and its density under standard conditions is approximately 0.0056. Consequently,

$$\text{Density of dry producer gas} = 0.0056 \times \frac{p \div 100}{2} = 0.000028p.$$

For the given data,

$$\begin{aligned} \text{Density of producer gas} &= 0.000028 \times 2576 \\ &= 0.0721 \text{ lbs. per cubic foot,} \end{aligned}$$

from which the weight of the dry producer gas per pound of dry coal is

$$W_g = 0.0721 \times 69.9 = 5.04 \text{ lbs.}$$

The volume of gas may also be obtained by metering it through the whole test, but the method outlined above will generally be found sufficiently accurate if reasonable care is taken in the sampling and analysis of the gas.

The weight of air supplied to the producer per pound of dry coal may be found from an expression similar to that for  $W_a$  for

coal combustion, page 179; the quantity  $D+M$  being replaced by  $c$ . The formula then is,

$$W_a = \frac{3.04 \times N_2 \times C_g}{c},$$

$N_2$  being the per cent by volume of nitrogen in the gas as given in column 2, Table 2. For the given data,

$$W_a = \frac{3.04 \times 56.4 \times 0.774}{33} = 4.02 \text{ lb.}$$

**The weight of steam supplied by the vaporizer per pound of dry coal is found by dividing the total weight of water to the vaporizer by total weight of dry coal. For the given data,**

$$W_s = \frac{268}{776} = 0.345 \text{ lb.}$$

Coming now to the heat balance, the first item is

**(b) Useful Heat in the Cool Gas.** This is exclusive of sensible heat if the gas is to be used for power, and is due to its calorific value when completely burned. The higher heat value will be used. Since all items of the heat balance are based on 1 lb. of coal, the useful heat may be found by multiplying the volume of gas per pound of coal,  $V$ , by the higher heat value in B.t.u. per standard cubic foot. The heat value may be figured as in column 7 of Table 2. This gives the heat values of the gas constituents, according to their amounts in one cubic foot, the sum of which is the required heat value. For the given data,

$$\text{Useful heat} = 69.9 \times 148 = 10,300 \text{ B.t.u.}$$

The heat value of the dry coal was found to be 13,040 B.t.u., so the useful heat is  $10,300 \div 13,040 = 79.0$  per cent. This is the "cold gas" efficiency of the producer.

(c) **Sensible Heat in the Dry Gas Leaving the Producer** is equal to

$$W_p \times \text{specific heat} \times (T_p - t).$$

The weight of the dry gas,  $W_p$ , has been previously figured.

The specific heat of producer gas, varies greatly with the hydrogen content. Except for hydrogen the average specific heat of all the gases, exclusive of steam, may be taken as 0.26

for a temperature of 1100° or 1200° F. The specific heat of hydrogen for these temperatures is about 3.72, so that even though its per cent by weight is always small the effect of hydrogen upon the average specific heat of producer gas is very marked and varies with the amount of hydrogen present. The curve, Fig. 91, gives

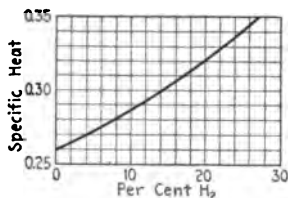


FIG. 91.—Specific Heat of Producer Gas.

the average specific heat of normal producer gas with various percentages of hydrogen. For the given data ( $H_2=10.4$  per cent) it is seen to be 0.288.

The temperature of the gas should be taken near the outlet at the producer with a suitable pyrometer.

For the given data,

$$\text{Sensible heat} = 5.04 \times 0.288 \times (1108 - 82) = 1490 \text{ B.t.u.}$$

or  $1490 \div 13,040 = 11.4$  per cent.

(d) **Heat Lost by Excess of Steam to the Producer.** This is estimated from the weight of steam found in the gas per pound of dry coal, and equals

$w_s \times \text{total heat per pound of the steam above feed temperature.}$

The total heat of the steam may be taken with sufficient accuracy for this calculation as

$$1000 + 0.5 \times T_p.$$

For a more exact value, the expression quoted on page 128 may be used.

To find  $w_s$  several methods may be used. It may be measured directly by drawing a sample of the producer gas through a tube of calcium chloride, the increase of weight of which gives the moisture absorbed. The gas is syphoned into a container of known size so that its volume, and hence its weight, may be obtained.  $w_s$  may also be figured from the hydrogen content of the coal and the gas. An easier method is based upon the fact that the sum of the weights of the gasified carbon, air, and water vapor fed into the producer must equal the weight of the producer gas including moisture. Hence,

$$C_g + W_a + W_s + m + \text{vapor brought in with air} - W_p = w_s.$$

The air vapor may be neglected here, although it may be a large percentage of the total steam, since its latent heat is not added by the coal. Then, for the given data,

$$w_s = 0.774 + 4.02 + 0.345 + 0.0283 - 5.04 = 0.137 \text{ lb.}$$

from which the heat lost to steam is

$$0.137 \times (1000 + 0.5 \times 1108) = 213 \text{ B.t.u.}$$

or

$$213 \div 13,040 = 1.6 \text{ per cent.}$$

(e) **Heat Transferred to Scrubber Water.** Items (c) and (d) together equal the heat given to the scrubber plus the heat radiated between the scrubber and producer outlets. For purposes of comparison, or to get an estimate of the amount of

sensible and latent heat in the gas without measuring it as just described, the heat added to the scrubber water may be determined. The water may be measured by a meter, and its temperature as for the jacket water of a gas engine. In the test quoted the total weight of scrubber water was 22,200 lbs. and its temperature rise was 45.8° F. Hence, the heat given to the scrubber per pound of dry coal is,

$$\frac{22,200}{776} \times 45.8 = 1310 \text{ B.t.u.}$$

**(f) The Heat Lost to Carbon in the Ash.** Taking the heat value of carbon as 14,600 B.t.u., this loss, per pound of dry coal is

$$14,600 \times C_a.$$

For the given data,

$$14,600 \times 0.042 = 613 \text{ B.t.u.}$$

or 
$$613 \div 13,040 = 4.7 \text{ per cent.}$$

**(g) Radiation from the Producer and other Losses.** These may be grouped in one item by subtracting the previously calculated heat balance quantities from the heat supplied, or some of them may be separately determined. The loss by leakage of air in the case of a suction producer plant may be found by comparing the useful heat, as herein calculated, before and after the dilution. In a pressure producer plant, any leakage is apt to be detected; the only way to measure it is by metering the gas since leakage does not alter its composition. The loss due to tar and soot may be found by a special analysis of the gas as it leaves the producer.

Grouping these losses for the given data, we have

$$\text{Radiation, etc.} = 100 - (79.0 + 11.4 + 1.6 + 4.7) = 3.3 \text{ per cent.}$$

(h) **Other Efficiencies and Results.** The "hot gas" efficiency is often expressed when the gas is used for heating. In this case the sensible heat is also "useful." If, then, the first three items of the heat balance in per cent are added, the result will be the hot gas efficiency. For the given data, this is

$$79.0 + 11.4 + 1.6 = 92.0 \text{ per cent.}$$

The "efficiency based on combustible," corresponding to the boiler efficiency so known, is the efficiency that would be obtained if there were no loss of fuel in the ash, and equals

$$\text{Eff. based on coal} \div \left(1 - \frac{C_a}{vm + fc}\right),$$

the quantity in the parenthesis being the efficiency of the grate, or cleaning. For the given data, this efficiency equals

$$79.0 \div \left(1 - \frac{.042}{0.869}\right) = 82.9 \text{ per cent}$$

for the cold gas.

For purposes of comparison of tests with different kinds of fuel, it is useful to obtain the **volume of gas per pound of combustible.** This equals  $V \div (vm + fc)$ .

In **capacity** tests, the output may be measured in cubic feet per hour by multiplying  $V$ , as herein calculated, by the weight of dry coal used per hour.

For **pressure producers** it is necessary to account for the coal used in the auxiliary boiler to make steam. This may be included in the total coal charged against the producer, and another item added to the heat balance to account for the heat lost in the steam making process. Or we may subtract from the volume of gas generated an amount equivalent to the steam supplied, which does not seem to be so logical a process.

For bituminous producers, if much carbon disappears as tar, it is necessary either to meter the gas to get  $V$ , or to determine the amount of carbon going to tar from 1 lb. of dry coal, by analysis of a sample extracted from a measured volume of gas. From this result a more correct value of the carbon gasified is obtainable.

The heat balance may be extended to cover the distribution of heat in a gas engine supplied by the producer, by the methods indicated under Test 53 covering this subject. In this case, the gas need not be metered since the fuel quantities are measured as coal, and the heat quantities should be based upon 1 lb. of dry coal instead of 1 cu. ft. of gas. The exhaust losses as calculated under Test 53 may be readily changed to the coal basis by multiplying them by  $V$  as herein obtained.

**Problem 55.** Using the data of the test example, what is the percentage of heat radiated from the scrubber and immediate piping?

**Problem 56.** Assuming that the engine of the example under Test 53 was operated with gas from the producer of the example of Test 55 with the same engine heat balance items, calculate these items in per cents of the heat value of the dry coal.

## 56. TEST OF A FAN BLOWER

**Principles.** There are two kinds of fan blowers called "pressure" and "volume"; the one delivering air at high pressure, the other at high velocity. The shape of the fan casing to a large extent determines its kind.

The capacity of a blower depends upon the volume of "free air" it will discharge in a given time at a given rotative speed. By free air is meant air at the pressure and temperature of the room at the time of the test. Capacity is generally expressed in cubic feet per minute.

The useful work done by a fan equals the energy imparted to the air as pressure and velocity. If  $W$  is the number of pounds



of air discharged per minute, and  $H_1$  is the pressure expressed as a head of air in feet, then  $WH_1$  is the foot-pounds of useful work per minute represented by the pressure energy. If  $V$  is the velocity in feet per second, then  $WV^2 \div 32.2$  is the foot-pounds of useful work represented by the kinetic energy. The quantity  $V^2 \div 32.2$  is the head of air, in feet, equivalent to the velocity, or "velocity head," which will be referred to as  $H$ . Then the useful, or "air horse-power," is

$$\text{A.h.p.} = \frac{WH_1 + WH}{33,000}.$$

This is referred to by the Air Machinery Code of the A.S.M.E. as the "gross" air horse-power; the difference between it and the "net" will be pointed out under Test 57.

**Selection of the Independent Variable.** In the operation of a fan blower, either the rotative speed, the velocity of air, or the pressure of the air may be varied. The first is controlled according to the type of the driving engine. The air velocity or pressure may be varied at a given rotative speed only by changing the size of the outlet from the air discharge pipe. A useful set of tests may be had by making the rotative speed the independent variable. The pressure is kept constant during a series of runs at different speeds; then the pressure is changed for another series, and so on. In order to keep the pressure constant when the speed is changed during each series, it is necessary to adjust the external resistance to the air. This may be done by using different nozzles or orifices as outlets from the discharge pipe. A convenient arrangement consists of a leaf shutter, similar to that of a camera, which may be readily changed to any desired diameter of outlet orifice.

A set of curves may be made between the various results and the rotative speeds. For each pressure there will be a corresponding set of curves. Taking the results from these curves

on a coordinate representing one value of the speed, another set of curves may be plotted with pressure as the independent variable. A number of such sets may be made corresponding to various speeds.

**The duration** of each run need not be more than a few minutes if a velocity meter is used for air measurement. The observed quantities vary very slightly, as a rule, during each run, so that only two or three sets of observations are needed.

(a) **Determination of Horse-power Supplied.** If the fan is belt driven, a transmission dynamometer may be used to advantage. Allowance should be made for belt losses, since the horse-power supplied to the fan shaft is desired. This may be done by using the revolutions per minute of the fan shaft with the torque shown by the dynamometer to calculate horse-power, and multiplying the result by the ratio of diameters of fan pulley to dynamometer pulley. This allows for belt slip.

If the fan is driven by an electric motor, the horse-power supplied may be had from a calibration of the motor, readings of the current supplied then being necessary.

(b) **Capacity.** For the measurement of the air quantity a meter of the velocity type is most readily adapted. Of these, anemometers (Test 26), orifices (Test 25), and pitot tubes (Test 23) are customarily used, preference being given to pitot tubes.

Velocities may be obtained from the pitot tube either by making a complete traverse of the pipe at each determination, or by locating the velocity opening at the point of mean velocity. For details, see Test 23 (a). Multiplying the velocities by the cross-sectional area of the pipe gives volumes of air under the pipe conditions of pressure and temperature. To reduce these results to free air, pressures and temperatures of the air in the pipe and room must be read. In the pipe, the pressure may be read from a manometer attached to a branch from the tube leading to the static opening of the pitot meter. For room pressure, the barometer should be read.

Consider as an example the following readings. Velocity head,  $h=0.5$  in. of water; pressure = 4 ins. of water; barometer = 29.9 ins. of mercury; temperature of room =  $65^{\circ}$  F.; temperature of air in pipe =  $67^{\circ}$  F. The absolute pressure of the air in the pipe is  $29.9+4\div 13.6=30.2$  ins. of mercury or  $30.2\times 0.49=14.8$  lbs. per square inch.

The absolute temperatures of the room and of the air in the pipe are  $525^{\circ}$  and  $527^{\circ}$ , respectively. The density of the air in the pipe may be figured from the familiar relation,

$$w = \frac{144p}{53.4T} = 2.7 \frac{p}{T},$$

which gives  $w = 2.7 \times 14.8 \div 527 = 0.0758$  lb. per cubic foot. To convert the velocity head  $h$ , in inches of water, into  $H$ , in feet of air, we have,

$$H = \frac{62.3}{0.0758} \times \frac{h}{12} = 68.5h,$$

for the given data. Consequently,  $H = 68.5 \times 0.5 = 34.25$ , and the velocity is

$$V = 8.02\sqrt{H}$$

$$V = 8.02\sqrt{34.25} = 47 \text{ ft. per second.}$$

If the area of the pipe is 0.33 sq. ft., the cubic feet of air per minute is  $60 \times 0.33 \times 47 = 930$ . To reduce this to free air, it is multiplied by the ratio of temperatures and the inverse ratio of pressures.

$$\text{Free air per minute} = 930 \times \frac{525}{527} \times \frac{30.2}{29.9} = 934 \text{ cu. ft.}$$

**(c) Horse-power Supplied per Thousand Cubic Feet of Free Air per Minute.** This quantity is obtained from the results of

(a) and (b) being the quotient between each two corresponding values divided by 1000.

(d) **Air Horse-power.** For the data previously given, the weight of air delivered per minute is  $930 \times 0.0758 = 70.5$  lbs. The velocity head is 34.25 ft. of air, and the pressure head,  $68.5 \times 4 = 274$  feet of air. Then the air horse-power is

$$\text{A.h.p.} = \frac{70.5(34.25 + 274)}{33,000} = 0.659.$$

(e) **Mechanical Efficiency Equals** the air horse-power divided by the horse-power supplied.

**Problem 56.** In the test of a blower, if gasoline, specific gravity = 0.75, is used as a gaging fluid with a pitot tube, what is the velocity head in feet of air, if  $h = 4.5$ ? Pressure is 6 ins. of the same fluid. Barometer is 30.3 ins. mercury. Temperature is  $70^\circ \text{F}$ . *Ans.*, 229 ft.

**Problem 56.** In the preceding how many foot-pounds of work will be done in one minute if pipe dia. = 6 in.? *Ans.*, 58100.

**Problem 56.** In the operation of a blower, the volume discharged varies very nearly as the rotative speed. The pressure is due to centrifugal force. From these facts, deduce how the power will vary with the speed.

## 57. TEST OF A RECIPROCATING AIR COMPRESSOR

**Principles.** In the operation of an air compressor, it is the purpose to increase the pressure of the air supplied so as to make available the energy it contains. As this energy is to be used when the air is cool, it is desirable to compress the air isothermally. If it is allowed to heat, the pressures during compression are higher and more work is required for compression. For this reason, water jackets and intercoolers are used, the heat removed by them being a saving.

Referring to Fig. 92, the dotted lines show an ideal air compressor diagram from a cylinder without clearance in which the air is discharged at a pressure,  $4_2 - 5_1$ , equal to that in the delivery

main. The supply is drawn in at atmospheric pressure along the line  $6_1-2_1$  and then compressed isothermally along  $2_1-4_2$ .

The actual indicator diagram varies from the ideal one as shown by the full lines, 1-2-4-5. The friction resisting the motion of the air through inlet and outlet valves and ports necessitates a lesser pressure than atmospheric to draw in the air, and a greater pressure than that in the delivery main to discharge it. Consequently, work is lost as represented by the areas 4-5<sub>1</sub>-5 and 1-2-3-6. The actual compression line, 2-4, must be above isothermal because of the impossibility of perfect cooling, and this results in the loss represented by the area between the lines 2-4 and 2-4<sub>1</sub>. These are "compression" losses.

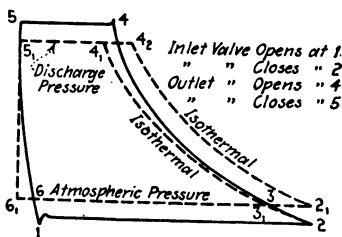


FIG. 92.—Ideal and Actual Air Compressor Diagrams.

When the outlet valve closes at point 5, an amount of air equal to the clearance volume is entrained in the cylinder. At the beginning of a new cycle, this air expands so that the effective stroke for drawing in air begins at point 6 instead of 6<sub>1</sub>. This point occurs later, the higher is the delivery pressure (at 5). When the inlet valve closes at 2, the air in the cylinder is rarified because of the suction, and it is not until 3 is reached that its pressure becomes atmospheric. The effective stroke for drawing in air is therefore represented by line 6-3 instead of the full length of the diagram. This is a "volumetric" loss, since less air is delivered than would be if the full stroke were effective. Another volumetric loss, not appearing in the diagram, is due to leaky valves and pistons, similar to slip of a pump.

After the air in the delivery pipe has been cooled down to room temperature, it contains energy, due to its pressure above

atmospheric, available for performing work. If this air could be expanded isothermally to its original pressure, there could be regained all of the energy necessary to compress it isothermally. This, however, cannot be done, the actual expansion being more nearly adiabatic. There is a final loss, then, due to the lack of availability of all the energy added to the air.

These various losses are indicated in certain expressions for efficiency, to be defined in the following, which are generally figured as air compressor test results. It is first necessary to define:

**Gross Air Horse-power.** This is the same as the indicated horse-power of a steam engine, being figured from the mean effective pressures of the indicator diagrams of the air cylinders.

**Net Air Horse-power.** This is the horse-power required to compress isothermally from room pressure and temperature to the pressure in the delivery main, a mass of air equal to that actually delivered.

**The Mechanical Efficiency** equals the gross air horse-power divided by the horse-power supplied to the compressor.

**The Volumetric Efficiency** is the cubic feet of free air actually delivered in a given time divided by the low pressure piston displacement in the same time.

**The Efficiency of Compression** is the net air horse-power divided by the gross air horse-power. This efficiency covers all the losses of the air cylinders due to faulty valve action, leakage, lack of cooling, etc., except that due to mechanical friction.

**Over-all Efficiency** is the horse-power available in the cooled compressed air if expanded adiabatically to room pressure divided by the horse-power available for running the compressor.

(a) **Capacity** may be expressed as the number of cubic feet of air discharged per minute at the pressure in the delivery main corrected to room temperature, or as the number of cubic feet discharged per minute reduced to free air, that is, air of the same temperature and pressure as that supplied to the compressor.

Let  $Q$  = cubic feet of air per minute discharged, as found by meter.

$T$  = absolute temperature, degrees Fahrenheit of air discharged.

$P$  = absolute pressure, pounds per square inch, air discharged.

$t$  = absolute temperature, degrees Fahrenheit, of air supplied (room).

$p$  = absolute pressure, pounds per square inch, air supplied.

Then the capacity in terms of compressed air is

$$Q_1 = Q \times \frac{t}{T},$$

and in terms of free air is

$$Q_2 = Q \times \frac{t}{T} \times \frac{P}{p} = Q_1 \times \frac{P}{p}.$$

$T$  and  $t$  are to be determined by thermometers, one in the room near the air inlet to the compressor; the other in the discharge pipe just beyond the last cylinder or receiver.  $p$  is obtained by a barometer and reduced to pounds per square inch;  $P$ , by a pressure gage set close to the thermometer measuring  $T$ .

The cubic feet of air  $Q$  flowing in the delivery main is a quantity difficult to measure accurately. The methods outlined under Part I are applicable, but, owing to the high pressures and large volumes, are not altogether satisfactory. Gasometers give the most reliable results. Two tanks similar to Fig. 53 may be connected together at the bottoms by a water pipe, and air piping arranged so that they may be alternately filled with the discharge from the compressor. As air enters one, water is displaced into the other until a mark on the gage glass is reached. The air is then diverted into the other tank, the

water returning to the first one which at the same time discharges the air it has just been filled with. The cross-sectional area of the tanks and the distance between upper and lower water levels, together with the number of fillings per minute, give the quantity sought. It is a good plan to pipe the air through a four-way cock so arranged that one motion connects one tank with the air delivery line and the other with the atmosphere.

(b) **Mechanical Efficiency.** The gross air horse-power is figured from the indicator diagrams from the air cylinders. The horse-power supplied to the air compressor, if it is steam driven, is the indicated horse-power of the steam cylinders. If it is belted or geared to the source of power, the horse-power supplied is that received at the compressor shaft, and should be obtained as for a fan blower, Test 56 (a). From these results the mechanical efficiency may be obtained.

(c) **Volumetric Efficiency.** The length of the line 6-3, Fig. 92, divided by the length of the diagram 6<sub>1</sub>-2<sub>1</sub> is often referred to as the "apparent" volumetric efficiency; apparent, because it does not take account of slip and leakage. The "true" volumetric efficiency is,

$$Q_2 \div L(A_h + A_p)N$$

in which  $L$  is the length of the stroke in feet,  $A_h$  and  $A_p$  are the areas in square feet of the piston at the head and power ends, respectively, and  $N$  is the number of double strokes per minute. If the compressor is single-acting,  $A_p$  should be omitted.

If the compressor has more than one cylinder, the expression applies to the low pressure cylinder.

(d) **Efficiency of Compression.** The net air horse-power must first be figured. The work done during isothermal compression of a gas is mathematically equal to

$$\text{abs. pressure} \times \text{volume} \times \text{hyperbolic log} \frac{\text{final abs. pressure}}{\text{initial abs. pressure}}$$



The volume to be considered in this case is that accounted for in the delivery main, corrected to room temperature. Then, using the notation given under (a),

$$\text{Net A.h.p.} = \frac{144PQ_1 \log_e \frac{P}{p}}{33,000}.$$

Dividing this by the gross air horse-power gives the required result.

(e) **Over-all Efficiency.** The horse-power available for running the compressor, in the case of a belted or similar machine, is the horse-power supplied to the shaft, the same as the denominator of the fraction expressing mechanical efficiency. For a steam driven machine, it is the horse-power available in the steam supplied. This may be taken as the horse-power that would be developed if the steam worked on the ideal Clausius cycle. If  $E_c$  is the Clausius cylinder efficiency, figured according to Test 42 (c), then

H.p. available in steam = I.h.p. in steam cylinder  $\div E_c$ .

The horse-power available in the air, if expanded adiabatically back to room pressure,

$$= \frac{\frac{144PQ_1}{0.41} \left\{ 1 - \left( \frac{p}{P} \right)^{.29} \right\}}{33,000}.$$

The numerator of this fraction is derived in works on thermodynamics and is in foot-pounds per minute, since  $Q_1$  is in cubic feet per minute.

Dividing the second horse-power expression by the first gives the over-all efficiency.

(f) **Separation of Losses.** A precise separation of all the losses cannot readily be made for the reason that they merge into each other. For example, mechanical friction in the air

cylinder adds heat to the air which makes the loss due to ineffective cooling greater. If there is leakage of air past the inlet valve during the compression, the compression curve will approach more nearly to the isothermal, making an apparent gain in the jacket's efficiency, but an actual loss as far as power and capacity are concerned. Again, if the inlet valve is mechanically operated and opens too early, the compressed air in the clearance space escapes and the apparent volumetric efficiency is improved, but power is lost.

If it is borne in mind that the losses as shown by the indicator diagram are not precise, an idea may be formed of them by an

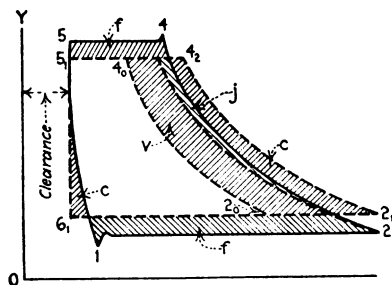


FIG. 93.—Air Compressor Losses.

analysis of a representative diagram from a test. Fig. 93 is such a one. Through  $2_1$ , vertically over 2, a horizontal should be drawn to represent atmospheric pressure. Through  $2_1$  an isothermal is drawn referred to a volume axis through 5. Through 2, another isothermal is drawn referred to the volume axis  $OY$ ,

located from the left of the diagram at a distance proportional to the clearance of the cylinder. Finally, a horizontal through  $4_2$  is laid off to represent the pressure in the delivery main. The ideal diagram is then  $6_1-2_1-4_2-5_1$ .

The loss due to fluid friction at the valves and ports is represented by areas  $f, f$ .

The loss due to insufficient cooling by the jacket water is represented by area  $j$ .

The loss in capacity due to volumetric inefficiency other than leakage is represented by the areas  $c, c$ . This is not a loss of power supplied, since the work is not done. It is merely

a diminution of the capacity of the compressor to deliver net horse-power.

The loss in capacity due to leakage is also a loss of power, because work is done on the air which leaks out of the system just as it is done on that which is discharged into the delivery main. To estimate this loss, a point  $2_0$ , Fig. 93, is located so that the distance  $6_1-2_0 = \text{distance } 6_1-2_1 \times \text{true volumetric efficiency}$ . Through  $2_0$  an isothermal is drawn referred to  $5-6_1$  as the volume axis. The diagram  $6_1-2_0-4_0-5_1$  then represents the net horse-power, and the area  $v$ , the loss of power due to leakage.

These areas may be integrated and expressed as per cents of the area representing the gross horse-power.

If the compressor has more than one cylinder, the indicator diagrams should be combined by reconstructing them on the same scale. The line  $4_2-5_1$  will then represent the receiver pressure. The next cylinder diagram will have this line as a datum in the same way that the low pressure cylinder diagram has the atmospheric line as a datum.

**(g) Heat Measurements.** The heat equivalent of the work delivered to the air piston

- = heat added to the jacket water
- + heat in the air in delivery main, above room temperature
- + heat radiated.

This follows from the fact that no energy is added to air when it is compressed, or expanded, isothermally, the isothermal being a "constant energy" condition. The heat removed per minute by the cooling water may be measured by taking weight and thermometer readings. The heat in the air may be figured from the data for the other results and from the specific heat of air. These quantities, considered with regard to the other results, will give a more complete idea of the efficiency of cooling. They

may be expressed in horse-power units by dividing by the number of heat units per minute equivalent to one horse-power, 42.42.

**Problem 57<sub>1</sub>.** An air compressor delivers 8.3 cu. ft. of air per minute, at 100 lbs. pressure absolute, and 220° F. What is its capacity in cubic feet of free air per minute; temperature of the room being 75°, and pressure, 29.5 ins. of mercury? What is its capacity in cubic feet of compressed air corrected to room temperature?  
*Ans.*, 45.1 and 6.52 cu. ft.

**Problem 57<sub>2</sub>.** The compressor of the preceding problem was two-stage, 14-in. and 9-in. bore by 12-in. stroke, and 100 working strokes per min. per cylinder. What is the volumetric efficiency from the same data? What would have been the volumetric efficiency if the compressor had been single stage, 14 in. × 12 in.? *Ans.*, 42.2%.

**Problem 57<sub>3</sub>.** What is the net air horse-power from data of Problem 57<sub>1</sub>? What is the horse-power available in the compressed, cooled air?  
*Ans.*, 5.41 H.p.

**Problem 57<sub>4</sub>.** How can the area *j*, Fig. 93, be roughly figured from heat measurements?

## 58. TEST OF A HYDRAULIC TURBINE

**Principles.** The horse-power delivered by a hydraulic turbine is as indicated by a dynamometer applied to its shaft. The horse-power available is that of the water which drives it, and is proportional to the difference in level between the head race and tail race, and the weight of water flowing per minute. The quotient between these horse-powers equals the mechanical efficiency.

When the turbine is supplied from a pipe line under pressure, as in the arrangement of wheels of the Pelton type, the available horse-power may be taken as that due to the pressure and velocity energy of the water just before entering the turbine. The machine is then not charged with the friction losses of the pipe line or with the losses due to its position in relation to the tail race.

**Selection of the Independent Variable.** In operation, any or all of the following may be varied. First, gate or needle valve opening; second, rotative speed; and third, brake horse-power. For test purposes, the head on the turbine may also be varied.

Change of the gate opening varies the amount of water supplied. Change of the brake horse-power is accompanied by a change of speed if the gate opening is left the same.

Laboratory tests are usually made at a constant head. The gate is adjusted at a predetermined opening for one series of tests. Then, by regulating the brake, the speed is varied for this series, and data obtained at each speed. Another series of tests is then made at a different gate opening, and so on until the full range has been covered.

If it is desired to test the performance at different heads, all or part of the above tests may be repeated after changing the head.

It is useful to plot brake horse-power and efficiency at each gate opening against speed.

**The duration** of each run need be only long enough to obtain accurate measurement of the rate of water supplied when the head remains practically constant, once uniformity of conditions has been established.

(a) **Available Horse-power.** This equals

$$\frac{WH}{33,000} \quad \text{or} \quad \frac{W(2.3p + V^2 \div 2g)}{33,000},$$

in which  $W$  is the weight of water per minute in pounds;  $H$ , the difference in level in feet, for the first case; and for the second,  $p$  is the pressure in the water pipe in pounds per square inch;  $V$ , the velocity in feet per second, and  $g$  the acceleration of gravity.

The most convenient method of measuring the water supplied is by weirs, the quantity usually being large. For small turbines, calibrated tanks may be used instead.

When the difference of level between the head race and tail race is measured, two datum marks may be made at the approximate positions of the upper and lower levels. The variations of the levels from these marks may be noted at a number of time intervals during each run. From these data, the average head on the turbine may be figured, the vertical distance between the marks having previously been measured.

When the second relation for available horse-power is used, the pressure may be obtained by a pressure gage just back of the needle valve. The velocity is readily obtained from the previously measured water quantity, the cross-sectional area of the pipe being known. Sometimes the velocity energy is small enough to be negligible.

(b) **Mechanical Efficiency** is obtained from the result (a) and the brake horse-power. The latter may be measured by one of the forms of friction brake.

(c) **Determination of Best Operating Speed.** The curve of efficiency vs. speed is something like an inverted U. The speed at the highest point of this curve gives maximum efficiency. It is approximately that speed which gives a peripheral velocity of one-half that of the jet for impulse wheels. For other types, it depends upon the characteristics of the turbines.

Under ideal conditions, the water upon leaving the turbine vanes would have no velocity except that due to the acceleration of gravity, since velocity of the off-flow means lost energy. With some types of turbine the best speed may be ascertained by watching the off-flow and determining that speed which is accompanied by the most nearly vertical descent of the water as observed by the eye.

**Problem 58.** What is the pressure energy in foot-pounds per minute available to a Pelton wheel supplied with 10 cu. ft. of water per minute at 100 lbs. pressure?  
*Ans.*, 143000 ft.-lbs.

**Problem 58.** In the preceding problem, if the pipe supplying the water is 2 ins. inside diameter, what is the velocity energy? What is the total horse-power?  
*Ans.*, 565 ft.-lbs. per min.; 4.35 H.p.

## 59. TEST OF A HYDRAULIC RAM

**Principles.** The hydraulic ram is a pump which uses the energy of a large volume of water under a low head in order to deliver a part of the water at an increased head. This is accomplished by establishing a flow through a waste valve at the foot of the supply pipe. When the velocity thus started attains a certain value, it closes the waste valve, whereupon the kinetic energy of the previously moving column in the supply pipe is converted into pressure great enough to open a valve into the discharge pipe. Water is then delivered at the increased pressure until the pressure energy is reduced to the static condition, when the discharge valve closes, the waste valve opens, and the cycle recommences. Fig. 94 shows diagrammatically the arrangement of piping of a hydraulic ram.

Two values for the useful work of a ram may be obtained, depending upon the level from which the head pumped against is dated. Referred to the level of the supply water, the useful work is that corresponding to the elevation of a weight of water through the distance  $H$ , Fig. 94.

Referred to the level of the ram, it is the work performed in lifting the same weight of water through the distance  $H_a$ . The one result leads to what is known as Rankine's efficiency, the other to D'Aubisson's. No confusion should arise through the existence of the two different efficiencies. Whether the one or the other should be used depends upon whether one is interested in pumping water from the supply level or from the level of the ram.

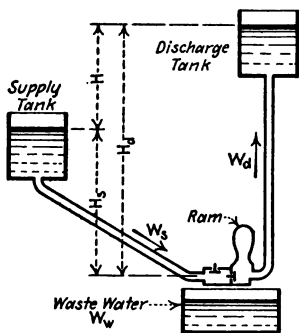


FIG. 94.—Hydraulic Ram.

**Selection of the Independent Variable.** Laboratory tests are generally conducted under a constant supply head, although this may be varied in the same way as for the test of a water turbine. If the supply head is maintained constant, either the number of strokes per minute of the waste valve or the discharge head may be made the independent variable.

(a) **Capacity** is expressed in gallons of water pumped per twenty-four hours. This may be measured by catching the discharge in a pail, or larger vessel if necessary, during a counted time. Instead of pumping against a static head as in regular operation, the discharge may be throttled by means of a valve in the discharge pipe which may then be short enough to collect conveniently the water delivered. The desired head is ascertained and regulated by the aid of a pressure gage between the ram and the throttle valve.

If the number of strokes per minute of the waste valve is the independent variable, it can be varied by changing the lift of the valve, or adjusting the spring tension if it is operated by a spring.

(b) **Efficiencies.** Let  $W$  and  $H$  stand for the weight of water flowing per minute in pounds and head in feet, respectively; and let the subscripts  $s$ ,  $w$ , and  $d$  refer to the supply, waste, and discharge, respectively, as indicated by Fig. 94. According to Rankine's efficiency a weight of water  $W_w$  must fall  $H_s$  ft. in order to raise  $W_d$  lbs. of water  $H$  ft. Therefore,

$$\text{Rankine's efficiency} = \frac{W_d H}{W_w H_s}.$$

According to D'Aubisson's efficiency, the energy available to the ram is the denominator of the following expression, and this energy accomplishes  $W_d H_d$  ft.-lbs. of work.

$$\text{D'Aubisson's efficiency} = \frac{W_d H_d}{W_s H_s}.$$



Since  $W_a$  was measured for the capacity determination, it is necessary only to measure the waste water to determine  $W_s$ . This may be done by allowing the waste to collect in a calibrated tank. The head,  $H$  may be found from the pressure gage readings, and  $H_s$  measured before the test. It is a good plan to use a float valve in the line which feeds the supply tank so that a constant level may be maintained in it automatically.

(c) **Curves** of capacity, total amount of water supplied in gallons per twenty-four hours, and efficiency against the independent variable may be plotted from the data previously mentioned.

**Problem 59.** The capacity of a ram operating against 20 lbs. pressure is 1042 gals. per twenty-four hours, and its efficiency (Rankine) is 35 per cent. The supply level is 10.5 feet above the ram. How much water passes through the waste valve in gallons per twenty-four hours?

*Ans.*, 10,100 gals.

**Problem 59.** Using the data of the preceding problem, what is the D'Aubisson efficiency?

*Ans.*, 45%.

## 60. TEST OF A CENTRIFUGAL PUMP

**Principles.** The useful work of a centrifugal pump is the product of a weight of water delivered in a given time in pounds and the total head pumped against in feet. The total head includes the friction heads of suction and discharge pipes and the velocity head if it is large enough to be of consequence.

The work supplied to the pump is that received by its shaft in the case of a belt driven machine, or one directly connected to an electric motor. In the case when the drive is a directly connected steam engine, both the drive and the pump are generally tested as a single unit as would be a reciprocating pump.

**Selection of the Independent Variable.** The rotative speed, quantity discharged, or head pumped against may be varied independently for a test. When one is varied arbitrarily, another is kept constant, and the third becomes a dependent variable. Two series of tests are useful; one at constant speed and the other

at constant head, the independent variable being head and speed, respectively. When the speed is kept constant, the head may be varied by throttling the discharge, and when the head is kept constant, the speed is varied by control of the motor driving the pump. In the latter case, each change of speed must be accompanied by a change in the valve opening in the discharge pipe in order to maintain a constant head.

(a) **Capacity** may be expressed in gallons per twenty-four hours or per minute. The usual methods of measuring water rates mentioned under the heads of hydraulic turbines and reciprocating pumps, may be used.

(b) **Horse-power supplied** may be measured as when testing a fan blower, Test 56 (a), except for steam driven pumps. In this case, the horse-power supplied is the indicated horse-power of the steam engine, the same as for a reciprocating steam pump.

(c) **Efficiency.** The horse-power delivered may be expressed in and found from the same terms expressing the horse-power available to a hydraulic turbine, Test 58 (a). Instead of measuring the distance between the level of the suction water and the gage in the discharge pipe, a vacuum gage may be placed in the suction pipe near the pump, and its indications translated into feet of head. The sum of this head and the vertical distance between the vacuum gage and the pressure gage in the discharge pipe should be added to the head indicated by the pressure gage; the result is the total head pumped against, neglecting the velocity head. The latter may be separately figured as for a hydraulic turbine.

Dividing the water horse-power by the horse-power supplied (b) gives the mechanical efficiency.

(d) **Curves** may be plotted, from the data obtained, as follows. For constant speed tests, power supplied, capacity, and efficiency against head. For constant head tests, the same results plotted against speed. Sometimes speed or head, power supplied, and efficiency are plotted against quantity discharged.

**Problem 60<sub>1</sub>.** How would you determine separately the work done by the various impellers of a stage centrifugal pump?

**Problem 60<sub>2</sub>.** Does or does not the measurement of total head by gages, outlined under (c), include the friction head of suction and discharge pipes, and why? Does it include the friction head of the water in passing through the pump, and ought this head to be included? Why?

## 61. TEST OF A HOIST

**Principles.** A hoist is a machine by which a large weight may be lifted by the application of a small force. This is accomplished usually by passing a chain, or chains, over a number of wheels geared together in such a way that a large motion of one end of the chain downward produces a small motion of the other end upward, whereby a mechanical advantage is obtained.

**The Ideal Mechanical Advantage** is the ratio of a distance moved through by the driving chain to the corresponding distance moved through by the following chain. Referring to Fig. 95, representing a hoist diagrammatically, this ratio is  $D/d$ . If there were no friction, this would be the ratio of the weight lifted,  $W$ , to the force applied  $F$ . As there is friction,

**The Actual Mechanical Advantage** equals the ratio of the weight lifted to the force applied, or  $W/F$ .

The efficiency of a hoist equals the work done in lifting the weight through any distance divided by the work done by the applied force acting through the corresponding distance.

If the efficiency of a hoist is more than 50 per cent, the weight lifted will return by gravity when the hoisting force is removed unless there is a locking device. Where this is provided, it is arranged to lock against the force of gravity only, and not against a force applied to the driving chain for the purpose of lowering the weight.

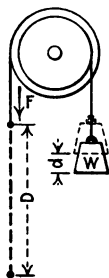


FIG. 95.  
Hoist.

(a) **The Ideal Mechanical Advantage** may be determined by actual measurement of the distances,  $D$  and  $d$ , Fig. 95. As  $d$  is generally very small compared with  $D$ , the result by this method may not have the desired degree of accuracy. A better method consists of counting the numbers of teeth of the various gears of the hoist and figuring from the data obtained, by the principles of kinematics, the velocity ratio of driver to follower.

(b) **The Actual Mechanical Advantage** may be determined by measuring the force applied to lift various loads. The desired results vary with the load, and may be plotted against its values. The loads may be applied by using various dead weights, sufficient to cover the working range of the hoist. The force required to lift these weights may be measured by applying the force through a spring balance hooked into the driving chain. Owing to the difficulty of reading a moving instrument, this is not a very accurate method. A better one is for the experimenter to stand on a platform scales and to apply the driving force at a uniform rate with one hand and at the same time, with the other hand, to balance the scales by adjusting the jockey weight. The weight of the experimenter minus the reading of the scales then equals the force applied to the hoist.

Several determinations of the force applied should be noted at each load, and their mean used to obtain the actual mechanical advantage at that load. It should not be attempted to make mental averages.

(c) **Efficiency** equals (see Fig. 95)

$$\frac{\text{Work done}}{\text{Work applied}} = \frac{W \times d}{F \times D} = \frac{\frac{W}{F}}{\frac{D}{d}} = \frac{\text{Actual mechanical advantage}}{\text{Ideal mechanical advantage}}$$

Consequently each result under (b) may be divided by that under (a) to get the corresponding efficiency. The efficiencies should be plotted against the loads.

**Problem 61.** Assuming that the friction losses of a hoist are constant throughout its working range, deduce the form of, and sketch roughly, curves between force applied and load lifted and between efficiency and load lifted. Will these curves pass through the origin or not, and why?

**Problem 61.** A differential hoist has 12 chain notches on the driving wheel and eleven on the smaller. What is its ideal mechanical advantage?

*Ans.*, 24.

**Problem 61.** The efficiency of a hoist is 60 per cent and its ideal mechanical advantage is 40. What load will a force of 4 lbs. lift? *Ans.*, 96 lbs.

**Problem 61.** Prove that a hoist having an efficiency greater than 50 per cent needs a locking device.

## 62. TESTS OF LUBRICATING OILS

**Principles.** The most indicative results of the relative values of lubricating oils are obtained from tests of viscosity, flash and chill points, and coefficient of friction.

**Viscosity** manifests itself as internal friction of the oil, a property which opposes the motion of the particles upon themselves, resulting in a reluctance to flow. It is related to, but is not proportional to the density. Since it opposes motion, it is an undesirable property, but there is such a thing as too little viscosity for good results. If the oil flows too readily, it may be squeezed out of the space between the surfaces it should lubricate, and the total amount of oil needed for proper lubrication might then become excessive. This depends largely upon the pressure between the bearing surfaces.

The term "body" is also used to denote viscosity.

**The flash point** is the temperature at which the oil will vaporize fast enough to form an explosive mixture with air. This temperature should be higher than the working temperature to be encountered in the service of the oil. Consequently it should be judged in connection with its service.

**The burning point** is the temperature at which a body of the oil considered will burn when a flame is placed a short distance over its surface.

**The chill point** is the temperature at which the oil loses its fluidity. Its determination is necessary only when the temperatures during service are low.

For more complete details concerning the requirements for and usual constants of lubricating oils, see Stillman's "Engineering Chemistry" or Kent's "Mechanical Engineer's Pocket-book."

(a) **The Specific Gravity** is of interest in connection with viscosity, since a high value of the one generally indicates a high value of the other. It may be determined by balancing a column of oil in one leg of a U-tube against a column of water in the other leg. Then the specific gravity of the oil equals the height of the water column divided by the height of the oil column. A more convenient method is to use a hydrometer if one is available.

Hydrometers are obtainable graduated to indicate specific gravity relative to the weight of water, but more usually they are graduated according to an arbitrary scale. The so-called Baumé scale is largely used in the United States, although the more logical scale based on the weight of water is preferable. Hydrometers so graduated indicate Baumé degrees (deg. Bé.). The relation between these degrees and specific gravity on the water basis is, for liquids heavier than water,

$$\text{Specific gravity} = 145 \div (145 - \text{deg. Bé.})$$

and for liquids lighter than water,

$$\text{Specific gravity} = 140 \div (130 + \text{deg. Bé.})$$

(b) **Viscosity.** This is a purely relative measurement. There are different standards for its expression, but, in general, it may be taken as the ratio of the time required by a given volume of the oil considered to flow through an orifice of a given size under a given head, or drop in head, to the time required by the

same volume of another liquid, as water, to flow under the same conditions.

The temperature of the oil has to do with its viscosity, so for comparative results the temperature must be kept constant during all tests. It should be remembered that the temperature during a test may be quite different from that in actual service, so the results are strictly comparative and not always indicative of the true merits of oils to be used at high temperatures.

Apparatus for viscosity tests, or "viscosimeters," may be readily improvised. It is desirable that the vessel containing the oil be water-jacketed to maintain a constant temperature. The orifice should be about  $\frac{1}{16}$  in. in diameter. The head on the orifice may start at about 6 ins. and end at about 4 ins., the total volume of oil passing through the orifice being that between the levels at the start and finish. Some forms of viscosimeters are arranged so that the flow shall be under a constant head.

**(c) Flash, Burning, and Chill Points.** The flash point may be found by placing the bulb of a mercury thermometer in a porcelain dish filled with oil to be tested, and heating the oil over a Bunsen burner until a flash may be obtained from it by passing a lighted taper over its surface. The temperature of the oil when this happens is noted as the flash point. The burning point is found similarly, by carrying the heating to a point at which the flash causes a burning of the oil in the dish. The chill point may be found by chilling a small amount of the oil in the bottom of a test tube by surrounding it with a freezing mixture until it congeals. The test tube is then removed and the oil stirred with a thermometer until it has warmed sufficiently to flow from one end of the tube to the other. The temperature is then noted as the chill point.

**(d) Coefficient of Friction.** To determine this, an oil testing machine should be used. See Test 33. The oil to be tested should be used under the same bearing pressure and the same

temperature as it will meet in service, as far as is possible. Comparative tests of different oils should be made under uniform conditions of temperature, pressure and velocity, or else the results will not be comparable.

(e) **Endurance Tests** have to do with the total amount of oil necessary to secure required lubrication. An oil that is satisfactory in all other respects may be of prohibitive cost because its lack of body may necessitate a large rate of feed. An idea of the endurance of an oil may be formed by supplying the bearing of an oil tester with a limited amount of it and noting the time required to raise the temperature of the bearing a predetermined amount. Another method is to note the time required to raise the coefficient of friction a predetermined amount. Still another is to measure the amount of oil during a given period of time when it is fed at a rate just sufficient to prevent a rise of temperature.

**Problem 62<sub>1</sub>.** What should be the least value of the flash point of an oil for a gas engine cylinder? Of an oil for a steam engine cylinder using steam at 100 lbs. absolute pressure? Of the same if the steam is superheated 200 degrees? Of an oil for a steam engine crank pin bearing?

**Problem 62<sub>2</sub>.** If the specific gravity of an oil is 0.85, what is its specific gravity in degrees Baumé? If specific gravity is 1.15, what is its value in degrees Baumé?

*Ans.*, 34.8°, 18.9°.



## APPENDIX

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### REPORTS OF ENGINEERING TESTS

A complete report should describe concisely the experimental object and how it was accomplished, and it should give numerical results and the conclusions formed from them. Whether the report is upon work performed by a student in the laboratory, or by an engineer in practice, its material may be arranged to advantage under the following heads.

- (1) Object of the test.
- (2) Description and principles of apparatus tested.
- (3) Method of testing.
- (4) Sample calculations.
- (5) Results and curves.
- (6) Discussion.
- (7) Rough notes or condensed observations.

The subjects under these sub-heads may be treated as follows.

**(1) Object of the Experiment.** This should be a clear, complete, and concise statement, preferably in one sentence. Its purpose, in practice, is to enable the reader to decide without a full reading whether or not the contents of the report come within his scope of interest. In student work, it should be included as a matter of training. In any case, the object of the test should be stated in writing before its performance in order that the experimenter and others concerned shall have a clear view of the undertaking.

**(2) Description and Principles.** This division should deal only with the machine, instrument, or apparatus tested. The extent of the treatment is governed by the requirements of those for whom the report is intended. A report prepared as a technical article or address kills itself if it talks over the heads of its audience or bores them with details with which they are familiar, however much satisfaction it may give the author. This is a matter for judgment. A good rule to follow is to give complete descriptions only of new or little known apparatus; for others it is sufficient to give only commercial sizes and names. It should be remembered, however, that any distinctive feature or any characteristic affecting the test results should be fully described. This depends upon the object of the test. For example, in the report of a mechanical efficiency test of a steam engine, it is appropriate to describe thoroughly anything affecting friction in operation such as lubrication details, balancing of valves, etc. But if the same engine were tested for its steam distribution, lubrication need not be mentioned at all; the valve mechanism then being the important item.

In this division there may be deduced any formulas used in getting results provided they are original or unusual. Otherwise, they may be quoted without deduction, with reference to the authority.

**(3) Method of Testing.** A logical way to begin this subject is by reference to the formulas for the quantities sought. Each formula, reduced to the desired form, shows the quantities that must be directly measured. The means of measuring them may then be described. Usual instruments may be merely mentioned by name, but special apparatus (original, or applicable only to the particular test) should be described fully. Any limitations of apparatus or unusual facilities affecting the precision of measurements should be mentioned to enable the reader to judge for himself the accuracy of the results. For

the same reason, the duration of the test should be stated with such items as frequency of readings. In this connection it is a good plan to refer to a sample set of observations which may be included in division 7.

**(4) Sample Calculations.** In practice, these should sometimes be included to make clear the method of testing and as a voucher of the accuracy of the calculations. They should be omitted if these points do not need amplifying. They should be brief, proceeding at once from the expression for the desired quantity, in which is substituted the test data, to the numerical result. The form under Test 56 (*d*) may be used as a model.

Reports by students should always contain sample calculations. The student should bear in mind that *sample* means something *representative of the whole*. Therefore, if five different quantities are tested for, three times for each one in the same way, five samples are necessary and sufficient. If these tests were repeated by another method involving a different calculation, ten samples are needed. If only one determination is made of each quantity, the sample must be all of the calculations to be representative.

**(5) Results and Curves.** Numerical results should always be presented in the form of curves when possible, and also as a table. This enables the busy reader to size up the report without fully reading it. Curves and tables should have definite titles and enough information to explain their meaning without reference to the text. Curve sheets should bear the scales of coordinates, and the axes should be scaled for ready reading. Where several curves appear on one sheet, their plotted points should be differentiated by such conventions as circles in outline, solid, half solid, etc. The points should be clearly marked so that they will not be obliterated when the curve is drawn in.

The student should take care to differentiate between *results* and *observations*. Sometimes observations are results, but this is seldom the case. Further, there may be many intermediary

quantities between the two which should not be confused with results. If the object of the experiment is clearly stated, it will always show what should comprise the results; in case of doubt, it should be referred to.

(6) **Discussion.** This should deal with the probable accuracy of the results as affected by the precision of the instruments and methods used and as indicated by the concordance of the results; it should compare the results with corresponding ones from similar tests, records of which are available in hand-books or elsewhere; and it should give the conclusions to be formed relative to the performance of the apparatus tested and to the physical laws controlling the performance. The conclusions to be formed from a test are the most important part of experimentation, in fact, its very *raison d'être*; so they should be fully given in the report. In practice, there would be no value whatever to a test from which conclusions could not be, or were not, made.

(7) **Rough Notes, Observations.** Complete reports should contain tables giving the observations in full so that their concordance and validity may be checked by the scientific investigator. When this is not considered necessary, a sample set may be submitted, as for division 3. Students should include in reports the original notes taken in the laboratory, a loose-leaf notebook being used for convenience.

Rough notes should be taken neatly and contain enough data not only to remind the experimenter of the noted quantities, but to enable an outsider without additional explanation to interpret them fully. Inexperienced observers, to save time, are prone to use arbitrary symbols or abbreviations in their notes, without meaning to anyone but themselves. The objections to this practice are that the notes cannot be checked by others and the observer himself is apt to forget their meaning.

In concluding the general subject of reports, it may be well to mention the custom of including in voluminous ones, as a sort of addendum, a synopsis of the whole work, relating briefly

the results and conclusions. This enables the busy reader to get the gist of it without a full reading. In technical papers addressed to scientific societies, or published as pamphlets, such a synopsis should conclude the report. If published as an article in a technical journal, it should be at the commencement.

## A METHOD FOR CONDUCTING STUDENT TESTS\*

The following method has been used with success at Syracuse University.

The first laboratory work assigned is, as far as the equipment will allow, individual. After a little training in the methods of the work, the experiments are given as problems, the theory of which is taught in the classroom. In the laboratory the student is quizzed at intervals during his work, to insure that it be performed not as a matter of rote. Approximate calculations from observations are exacted as the observations are obtained. It is a great mistake to allow the postponement of such calculations until after the test is completed. This point cannot be too strongly emphasized. It is far better to have an experiment only partly but intelligently done than to have a vast amount of observations leading to faulty results and a half-grasp of the principles.

As the course advances, the tests, especially those upon large units, require more than one student to make all the necessary operations and measurements. For the successful conduct of some, it is expedient to have four or five stations at each of which one observer is needed. These advanced tests are undertaken in this way. In the classroom the students solve problems upon the principles involved and are given the special instruc-

\*Abstracted from a paper by the present writer on "The Teaching of Experimental Engineering," *Educational Review*, June, 1912.

tions which they could not reasonably be expected to obtain for themselves. When they have shown a satisfactory understanding of the principles, they are allowed to proceed with the experimental work, conducted as follows: For a test requiring five stations, for instance, a squad of six men is selected. For the first fifteen minutes or so of the test, observers A and B are at station No. 1, C at No. 2, D at No. 3, E at No. 4, and F at No. 5. Then B moves to station No. 2, where he is instructed by C in its duties. As soon as B has become familiar with them, C moves on to station No. 3 and is instructed by D concerning the work there, after which D moves to station No. 4 with E, and so on. The scheme can best be understood by the following schedule.

## STATION NO.

TIME	1	2	3	4	5
10 : 00- : 15	AB	C	D	E	F
: 15- : 20	A	BC	D	E	F
: 20- : 25	A	B	CD	E	F
: 25- : 30	A	B	C	DE	F
: 30- : 35	A	B	C	D	EF
: 35- : 40	FA	B	C	D	E
: 40- : 45	F	AB	C	D	E
	Etc.				

The advantages of this method over shifting all the students at one time are obvious. There is absolutely no break in the accuracy of the observations or continuity of the test since only one man shifts at a time and he does not make measurements to be used before he has become somewhat familiar with the apparatus. The instructor is relieved of the small but essential details of instruction by the students themselves. Room is made for an additional man on the test without sacrificing the work of any of the others. A complete shift of every man can be made in a shorter time; in the example cited, the interval is thirty minutes.

The time necessary for an observer at a station to instruct the newcomer varies, of course, with the duties of the station. In a boiler test the average time is about ten minutes, so that a complete shift of six men would be effected in sixty minutes. If all the men shifted at once, the interval should not be less than ninety minutes, preferably two hours.

The shifting is automatic. All that each student needs to know is the sequence of stations, and to remember that he is to move to the next one only when he has been relieved by, and has instructed his successor.

It has been found of considerable advantage to include a station the duty at which is to maintain a "General Log." This contains the observations from all the stations. The student in charge of it checks to some extent these observations upon recording them, and is required also to calculate roughly indicative results as the test proceeds. The log may be used for reference by all the students and enables a clearer view of the whole test.

At the end of a complete shift, three of the men are replaced by three new ones, and put upon the final calculations under the instructor's supervision. When these are completed the resulting quantities are plotted on a large chart, in common use for all who have made the test. In the meantime the test is continued by the three new and the three old observers, the arbitrarily varied quantity of the test having been changed. At the end of the next complete shift, half the men are again replaced, the ones remaining being the more recent ones. In this way each group of three men serves two complete shifts and makes the calculations from the observations of one. When all the results are in, their concordance is checked by the regularity of the plotted curves, and these are presented as a whole to the class for consideration.





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